Public Works Canada
Solar Programs Office

SOLAR DESIGN COURSE MANUAL

Part I: FUNDAMENTALS

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SOLAR DESIGN COURSE MANUAL

PART 1 - FUNDAMENTALS

FUNDAMENTALS
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CHAPTER 1

INTRODUCTION

1.1 COURSE CONTENT

This course in Solar Energy System Design has as its objective the training of engineers and others who will be involved in Public Works Canada's PUSH program. It is directed towards both the private and the public sectors and towards staff who will be supervising and administering PUSH projects as well as towards design engineers.

The course will be presented in two parts. Part I, "Fundamentals", will cover solar radiation, collector heat transfer, an introduction to the design of liquid and air systems, collector durability, and system components. Part II, on "Advanced Topics", will cover design methodology, site selection, simulation techniques, design approach, and inspection and operational check-out. A major element of the course on "Advanced Topics will be a practice session on the use of F-CHART 4.0 through a computer terminal.

The course is offered in two independent segments, for the benefit of those who may already be familiar with solar energy fundamentals, or who do not have time to take the whole course at once.

The design of space heating systems and of domestic and process water heating systems will be dealt with, but the emphasis will be on the water heating systems since they are the more common, as well as the more economic, applications of solar energy in Canada.

1.2 CLASSIFICATION OF COLLECTORS AND SYSTEMS

Solar collectors may be either air cooled or liquid cooled. This provides the major distinction not only between collectors, but between system types. Air cooled collectors are normally glazed flat plates. Liquid cooled flat plate collectors may be glazed or unglazed. Another type of liquid cooled collector is the evacuated tube, in which convective heat loss is prevented by an evacuated annulus around the absorber tube. There are many other types of liquid cooled collectors with various concentration arrangements, but no concentrating collector is approved under the PUSH program.

Apart from collector type, the major distinctions between solar energy systems are between active and passive systems, between applications of the solar energy collected, and between methods of freeze protection. Only active systems are elegible for installation under the PUSH Program, and only active systems will be dealt with in this course. The major applications of solar energy in Canada are:

1. swimming pool heating
2. domestic and commercial service water heating
3. industrial process water heating
4. make-up air preheating
5. space heating
6. drying of agricultural crops or other materials.

Solar cooling is also possible in Canada, although it is unlikely to be economic.

Of the above applications, drying and air preheating are best handled by air systems, space heating is usually best handled by air systems, and the other three applications are best handled by liquid systems.

Air systems need no freeze protection, of course. Some liquid systems, particularly swimming pool heaters, use the simplest of all freeze protection systems, shut-down and drainage over the winter. For those systems which must operate in freezing weather, freeze protection can be provided by using a liquid which does not freeze at ambient conditions (such as a glycol solution or an organic liquid) to cool the collectors, or by automatic drainage of the collectors when freezing is a possibility. If the latter technique is used the collectors may "drain back" into a storage tank or "drain down" out of the system. In some climates, recirculation of warm water from storage is used to prevent freezing, but this is not feasible in Canada's cold winters.

1.3 HISTORICAL REVIEW

The concept of solar heating has been well known since the beginning of the century and, before natural gas became available in the late 1940's, solar domestic water heaters were common in Florida and Southern California. They have been used for many years in Australia and Israel. Early research and development work on solar space heating was done at MIT in the 1940's and 1950's, but it was not until the sharp rise in oil prices in the early 1970's that there was renewed interest and activity in solar energy in North America.

The first active solar space heating system in Canada was built by Erich Hoffman in Surrey, B.C. His system consisted of 37 m^2 of collector with 3600 L of heat storage in tanks in a small room in the basement. To heat this house, a door is opened to allow the circulation of warmed air.

About 1975 the Canadian Government began supporting solar heating in Canada. Greg Allan's Gananoque house received a $15,000 grant. It has 22 m^2 of collectors with heat storage in 1800 kg of wax in plastic bags. Another Government supported project was the Meadowvale house at Mississauga, Ontario. This house, with 125 m^2 of floor area, has 64 m^2 of liquid cooled collectors and heat storage in a 22730 L concrete water tank.

Not all solar energy projects of this period were Government supported. Martin Stanley-Jones built a highly energy conserving 172 m^2 house near Ottawa, and heated it with 19 m^2 of Sunworks collector, storing heat in 22700 L of water. Rick Symmes built a plug-on air collector with pebble bed storage, which cost him $1,300 in scavenged material and which he estimated would meet 25% of his heating load.

Two ambitious Government funded projects of the late 1970's were the new Alchemy Ark near Charlottetown and Provident House. In the former, 79 m^2 of water cooled collector provided heat to 95,500 L of storage, for space heating. In the latter, the concept of annual storage was applied for the first time in Canada. Provident House has 260 m^2 of floor area and is heated by 67 m^2 of water cooled collector panel. Heat storage is in a 273,000 L tank, filling the basement, which is designed to store heat from the summer to provide 100% of the winter heating needs.

The Canadian Government concluded, in 1978, that the approach they had taken to supporting solar energy was not leading to the development of a Canadian Industry. Therefore they set up the Solar Programs Office within the Public Works Canada, and funded the PASEM (Program of Assistance to Solar Equipment Manufacturers) and PUSH (Purchase and Use of Solar Heating) programs. PASEM, with a budget of $4.1 million, funded the development of 21 new solar products by ten Canadian solar equipment manufacturers. PUSH with a budget of $125 million, aims to use this equipment to provide space and water heating to Federal Government buildings. About 500 projects had been funded by 1981, ranging from domestic hot water systems up to systems with over 500 m^2 of collector costing over a quarter million dollars.

Another major solar energy program currently underway in Canada is the Energy, Mines and Resources Domestic Hot Water (DHW) Demonstration Program, which is installing about 900 residential solar DHW systems across Canada.

Present activity in the Canadian solar industry is concentrated in these Federal Government programs, and in several program funded by provincial governments. At present, private installations of solar energy systems are mostly of swimming pool heaters, which offer the purchaser a reasonably short payback period at today's escalating energy prices.

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CHAPTER 2

SOLAR RADIATION

2.1 INTRODUCTION

The aspects of solar radiation covered in this chapter are:

1. the physical nature of solar radiation,
2. the geometry of solar radiation,
3. the availability of solar radiation data, and
4. the conversion of data to the required form.

A more thorough discussion of this material is covered in Chapters 1, 2 and 3 of "Solar Engineering of Thermal Processes" by Duffie and Beckman (D and B). For clarity, the nomenclature used here is that used by D and B.

2.2 THE NATURE OF SOLAR RADIATION

Thermal radiation is electromagnetic, like x-rays, radio, gamma radiation, etc., and follows the wave equation

$$C = C_0/n = \lambda \upsilon \qquad \text{(Eqn 2.2.1)}$$

The basic radiation properties of a material object are:

1. absorptance $\alpha = q(\text{absorbed})/q(\text{incident})$ (Eqn 2.2.2)
2. reflectance $\rho = q(\text{reflected})/q(\text{incident})$ (Eqn 2.2.3)
3. transmittance $\tau = q(\text{transmitted})/q(\text{incident})$ (Eqn 2.2.4)

The words transmittance, absorbtance and reflectance are used rather than transmissivity, absorbtivity and reflectivity because these are properties of specific objects rather than of particular materials. A pane of glass of a particular thickness will have a transmittance depending on its thickness, while the transmissivity is a property of the material of which the pane is made and is independent of thickness.

It might seem that the words reflectance and reflectivity should be interchangable since the phenomenon they measure is independent of the geometry of the body behind the surface. However, surface roughness, oxide coatings, etc., change the reflectance of surfaces, so the word reflectivity is reserved for idealised smooth, clean surfaces, so that it depends only on the surface material, while reflectance is applied to real surfaces.

A surface is called black if it can absorb all incident radiation for all wavelengths and for all angles of incidence; that is, it has an absorbtance of 1. A black body at the absolute temperature T will emit an amount of energy per unit area equal to

$$E_b = \sigma T^4 \qquad \text{(Eqn 2.2.5)}$$

where is the Stefan Boltzman constant.

$$\sigma = .1714 \times 10^{-8} \text{ BTU/ft}^2 \text{ hr } {}^0R^4$$

$$= 5.6697 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$$

No surface can emit more energy per unit area than a black body at the same temperature. Emittance is defined as

$$\varepsilon(T) = E(T) / E_b(T) \qquad \text{(Eqn 2.2.6)}$$

Note that emittance is a function of temperature, as are the other radiation properties.

Kirchoff's law states that $\varepsilon_\lambda(T) = \alpha_\lambda(T)$ for any surface. However, this does not imply that $\varepsilon = \alpha$ over the whole range of wavelengths for any particular temperature. Thus it is possible to produce a selective surface, which has an absorbtance of 0.9 for solar radiation, concentrated in the wavelength range from 0.2 to 1.0 microns, and has an emittance of 0.1 when it is at 300 K and is emitting radiation in the range from 7 to 15 microns. The wavelength distribution of blackbody radiation is Planck's law:

$$E_{b\lambda} = \frac{2\pi h C_o^2}{\lambda^5 (e^{hC_o/\lambda kT} - 1)} \qquad \text{(Eqn 2.2.7)}$$

where h is Planck's constant

k is Boltzman's constant

$C_1 = 2\pi h C_0^2 = 3.7405 \times 10^{-16}$ W m²

$C_2 = hC_0/k = .0143879$ m K

This distribution is plotted on a log scale for three temperatures in Figure 2.2.1.

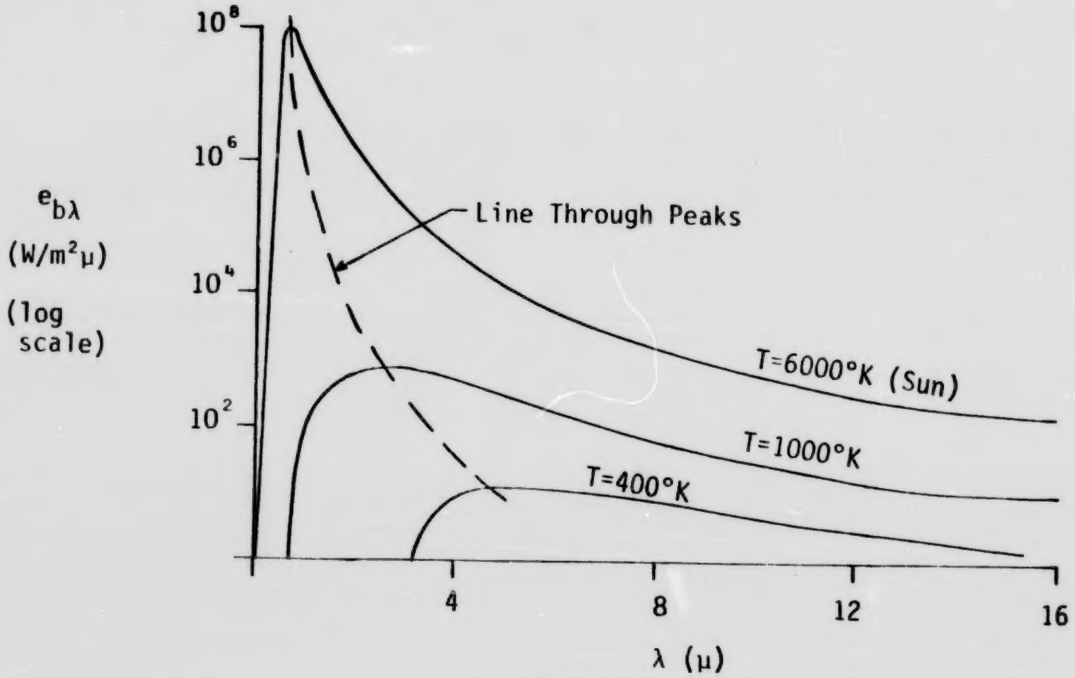


FIGURE 2.2.1

DISTRIBUTION OF EMISSIVE POWER WITH WAVELENGTH

Wien's displacement law

$$\lambda_{max} T = 2897.8 \mu K \qquad \text{(Eqn 2.2.8)}$$

describes the line throughout the peaks.

A normalized plot of these three curves is shown in Figure 2.2.2.

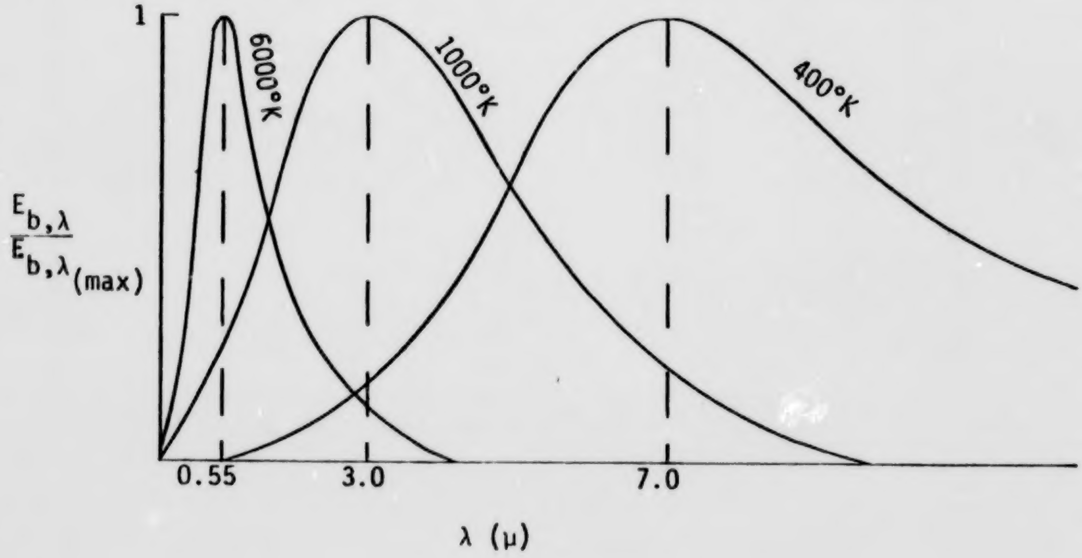


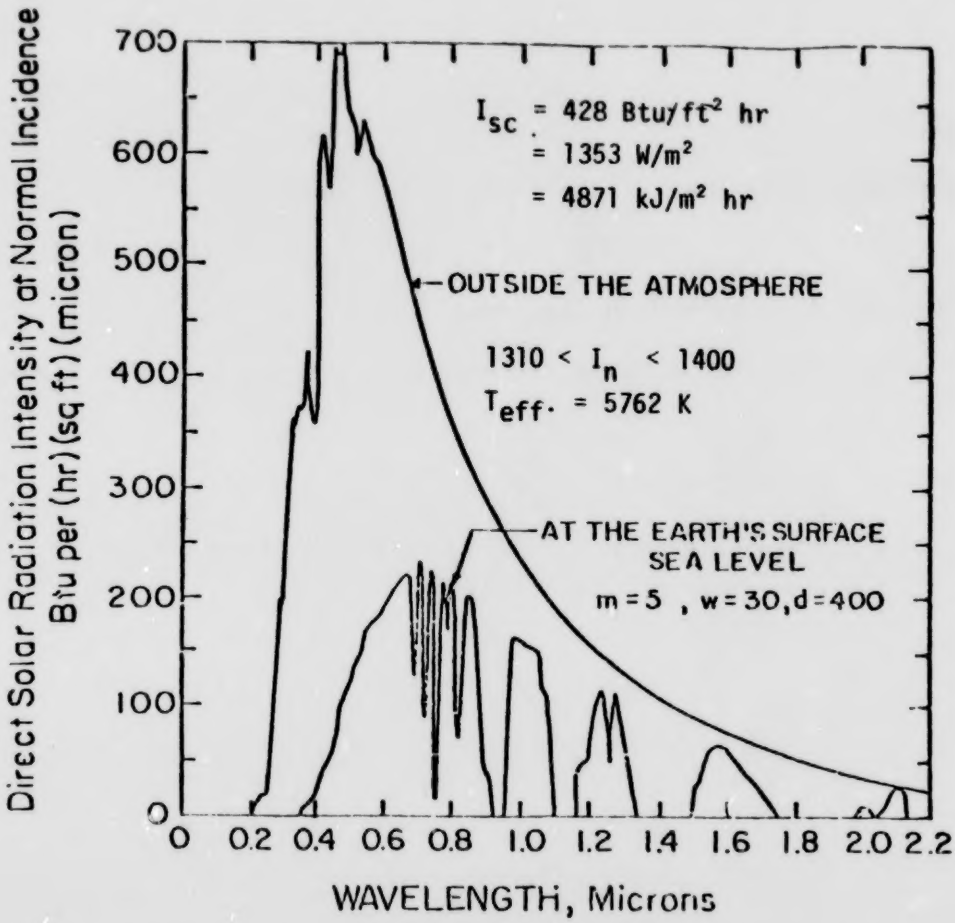
FIGURE 2.2.2

NORMALIZED DISTRIBUTION OF EMISSIVE POWER
AS A FUNCTION OF WAVELENGTH

Integration of the equation for $E_{b\lambda}$ yields the areas under these curves

$$E_b = {}_0\int^{\infty} E_{b\lambda} d\lambda = \sigma T^4 \qquad \text{(Eqn 2.2.9)}$$

No real surface emits as a black body. The sun's emission approximates that of a black body at 5762 K, but, as Figure 2.2.3 shows, it is not completely smooth. By the time the radiation penetrates to the earth's surface its frequency distribution has been greatly influenced by absorbtion by ozone, water vapour and carbon dioxide and by dust particles.



SPECTRAL DISTRIBUTION OF DIRECT SOLAR RADIATION AT NORMAL INCIDENCE FOR THE UPPER LIMIT OF THE ATMOSPHERE AND AT THE EARTH'S SURFACE FOR CONDITIONS AT SEA LEVEL, WHERE :

$$m \equiv \text{Air mass} = \frac{\text{length of path of sun rays through the atmosphere}}{\text{shortest path of sun rays through the atmosphere}}$$

w ≡ Depth of atmospheric precipital water — mm

d ≡ Dust particles per cc

m = 5 ≡ The suns altitude of 11° and Transmissivity ÷ 0.28.

RADIATION SPECTRUM

FIGURE 2.2.3

If there were no atmosphere the solar intensity at the earth's surface would be 1353 w/m^2 with a variation of ±3% depending on the time of year. This is because the earth's distance from the sun varies through the year, so that the solar intensity must also vary according to the inverse square law. Where no measured data is available, it is possible to estimate the clear sky radiation from the solar constant, using the method outlined in Section 2.8 of D and B.

2.3 THE GEOMETRY OF SOLAR RADIATION

Knowing the strength of the sun's beam at the earth's surface does not provide sufficient information to calculate the energy falling on a collector. This is because

1. the collector is not normal to the beam, and
2. much of the radiation falling on the collector is diffuse radiation from the sky and the ground.

The geometric problem is the simpler of the two, depending only on mathematical analysis. Figure 2.3.1 shows the nomenclature for this analysis.

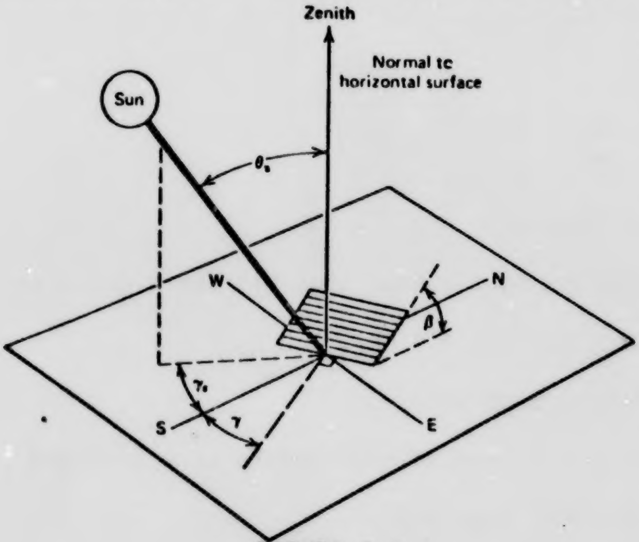


FIGURE 2.3.1

ZENITH ANGLE, SLOPE, SURFACE AZIMUTH ANGLE AND SOLAR AZIMUTH ANGLE FOR A TILTED SURFACE

The equation for the angle of incidence, Θ, of the solar beam on a surface is:

$$\begin{aligned}\text{Cos } \Theta = {} & \sin\delta \sin\phi \text{ Cos } \beta \\ & - \sin\delta \text{ Cos } \phi \text{ Sin } \beta \text{ Cos } \gamma \\ & + \cos\delta \cos\phi \text{ Cos } \beta \text{ Cos } \omega \\ & + \cos\delta \sin\phi \sin\beta \text{ Cos } \gamma \text{ Cos } \omega \\ & + \cos\delta \sin\beta \sin\gamma \text{ Sin } \omega \end{aligned} \qquad \text{(Eqn 2.3.1)}$$

where ϕ = latitude (N = +)

δ = declination (N = +)

β = slope of surface (horizontal = 0)

γ = surface azimuth (S=0, W = +90°)

ω = hour angle (11:00 = -15°
13:00 = +15°)

The declination is given by

$$\delta = 23.45° \text{ (Sin } 360)\left(\frac{284 + n}{365}\right) \qquad \text{(Eqn 2.3.2)}$$

where n = day of the year.

The time for determining the hour angle is sun time, given by

$$\tau = \text{standard time} + E + 4\left(L_{ST} - L_{LOC}\right)$$

where E = the equation of time

L_{ST} = the longitude on which standard time is based

L_{LOC} = the local longitude

The equation of time is shown in Figure 2.3.2, and is given by

$E = 9.87 \text{ Sin } (2B) - 7.53 \cos(B) - 1.5 \text{ Sin } (B)$ (Eqn 2.3.3)

where $B = 360\ (n-81)/364$

and n = day of the year.

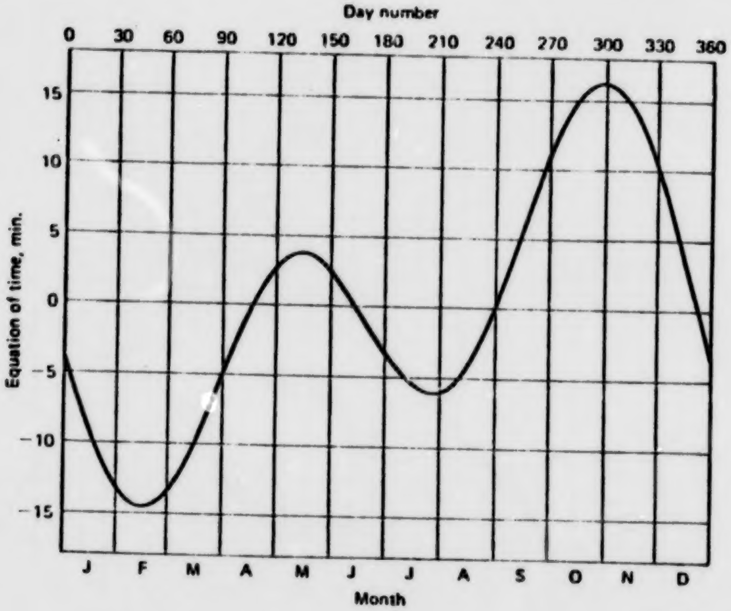
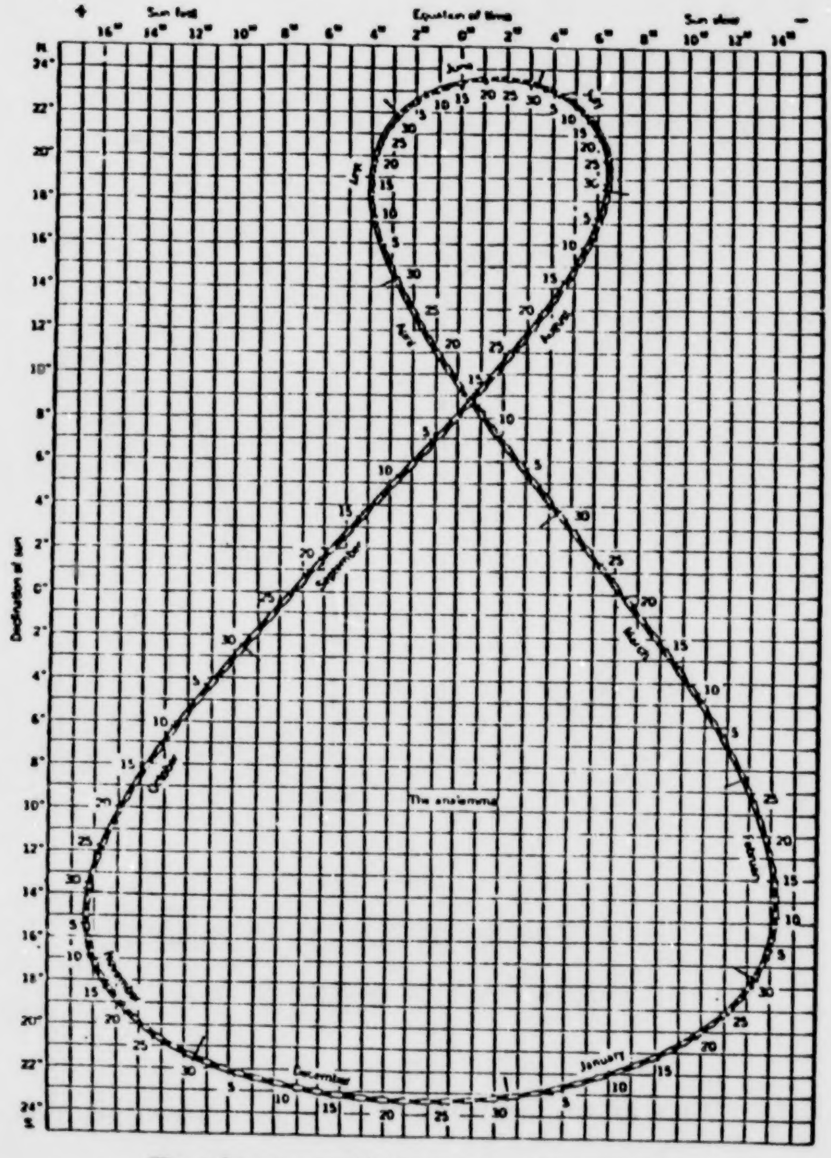


FIGURE 2.3.2

THE EQUATION OF TIME, E, IN MINUTES,

AS A FUNCTION OF TIME OF YEAR

Another form of the equation of time is a graphical presentation known as the analemma. The analemma, Figure 2.3.3, can also be used to determine the declination of the sun.



The analemma – a graphical presentation of the equation of time. Equation of time values are plotted to the right and left of the vertical centerline.

FIGURE 2.3.3

THE ANALEMMA

Figure 2.3.4 and Figure 2.3.5 illustrate the effects of collector tilt on the amount of radiation intercepted.

By applying Equation 2.3.5 and the solar intensity one can find the radiation on a horizontal surface outside the earth's atmosphere on an average day in each month. This is presented in graphical form in Figures 2.3.6 and 2.3.7 and in numerical form in Table 2.3.1.

A study of these curves makes it apparent that the north pole receives more horizontal radiation than any other area of the globe in midsummer, even though the sun is higher further south. This raises the question of how to take advantage of this fact while using a flat-plate collector.

Table 2.3.1 will be of interest later since it contains data which will be needed in handling the problem of eliminating diffuse radiation.

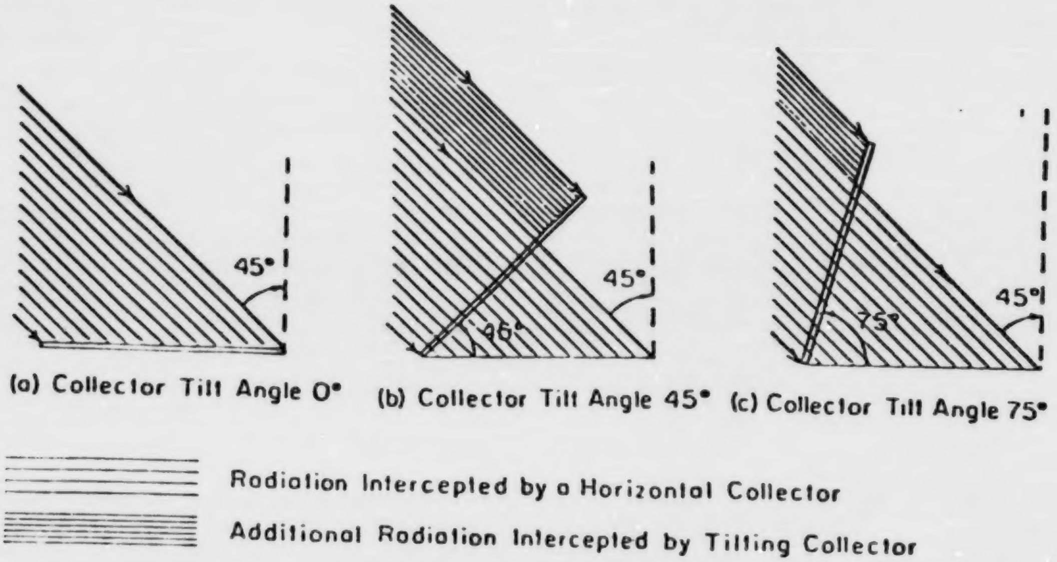
R_b, a geometric factor, is the ratio of the beam radiation on a tilted surface to that on a horizontal surface at the same instant.

$$R_b = \frac{G_{bt}}{G_b} = \frac{\text{Cos } \Theta_T}{\text{Cos } \Theta_z} \qquad \text{(Eqn 2.3.4)}$$

Using Equation 2.3.1,

$$R_b = \frac{\text{Cos } (\phi-\beta) \text{ Cos } \delta \text{ Cos } \omega + \text{Sin } (\phi+\beta) \text{ Sin } \delta}{\text{Cos } \phi \text{ Cos } \delta \text{ Cos } \omega + \text{Sin } \phi \text{ Sin } \delta}$$

(Eqn 2.3.5)


45°
45°
45°
45°
75°
45°
Radiation Intercepted by a Horizontal Collector
Additional Radiation Intercepted by Tilting Collector

(a) Collector Tilt Angle 0° (b) Collector Tilt Angle 45° (c) Collector Tilt Angle 75°

FIGURE 2.3.4

EFFECT OF TILTING THE COLLECTOR
ON ENERGY INTERCEPTED

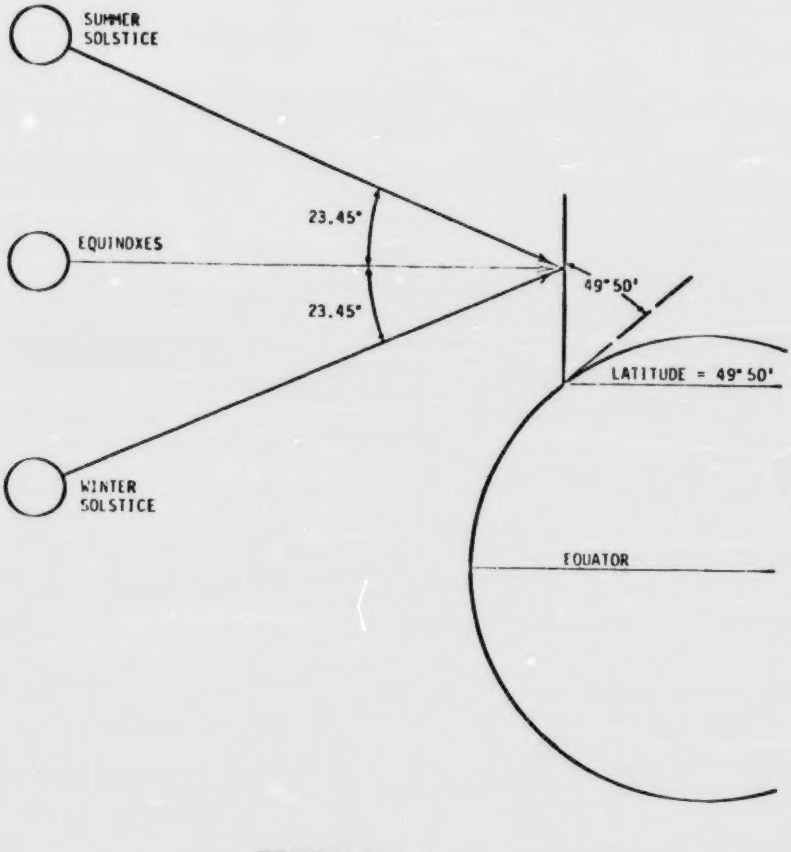
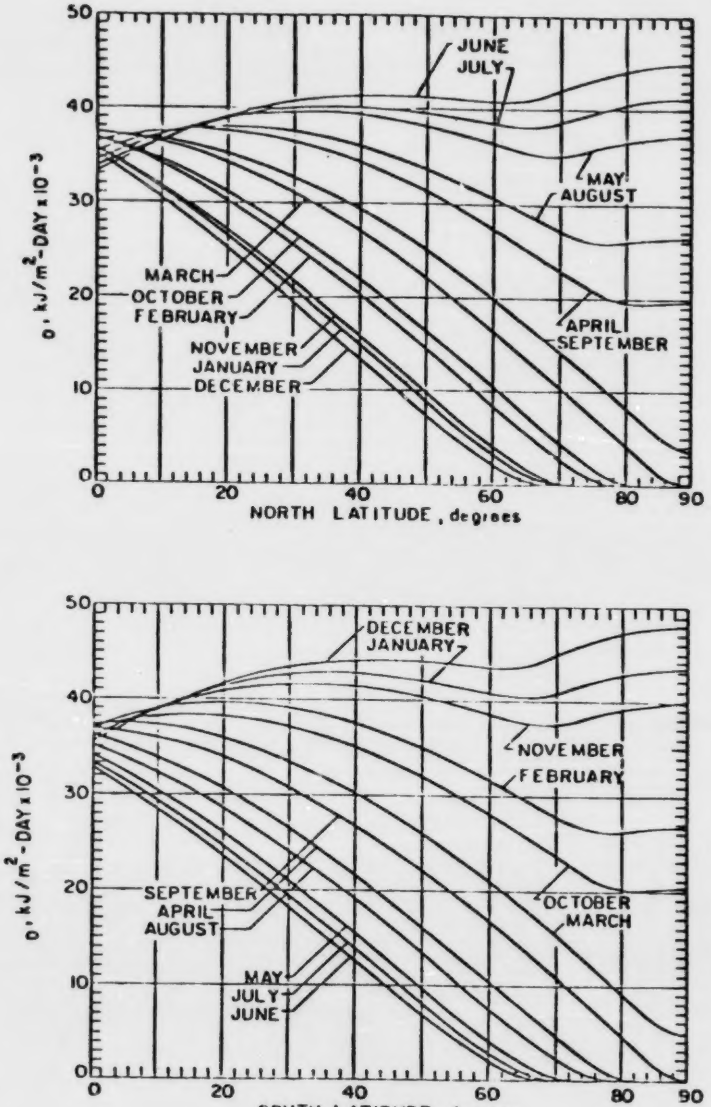

SUMMER
SOLSTICE
EQUINOXES
WINTER
SOLSTICE
23.45°
23.45°
49° 50'
LATITUDE = 49° 50'
EQUATOR

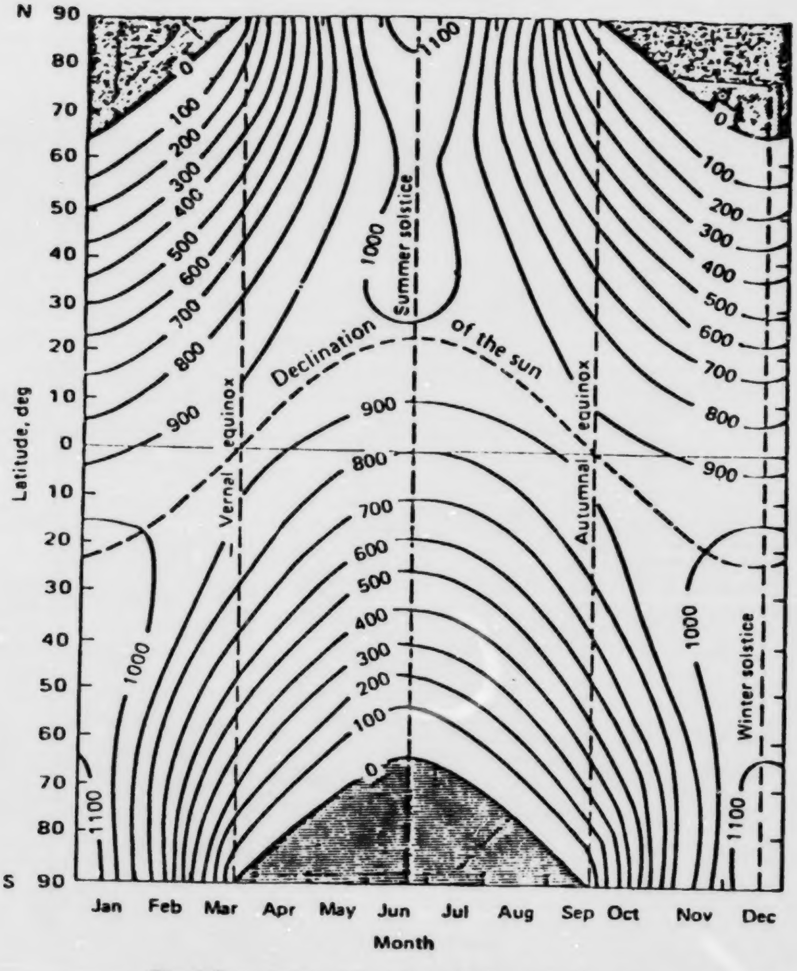
FIGURE 2.3.5

INCIDENT ANGLES ON A COLLECTOR
TILTED AT AN ANGLE EQUAL TO THE LATITUDE



Extraterrestrial daily insolation on a horizontal surface, for mid points of months.

FIGURE 2.3.6



The daily variation of the solar radiation at the top of the atmosphere as a function of latitude. The values are measured in langleys per day. (1 langley = 1 cal/cm^2 = 3.687 Btu/ft^2 = 11.6 W · hr/m^2.) From Kreider, J. F., and F. Kreith, "Solar Heating and Cooling," revised 1st ed., Hemisphere Publ. Corp., Washington, D.C., 1977.

FIGURE 2.3.7

Table 2.3.1

$\overline{I}_0$, Monthly Average Daily Extraterrestrial Radiation on a Horizontal Surface
GJ/m^2 day

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20	26.7	30.4	34.4	37.6	38.9	39.2	39.0	37.9	35.3	31.4	27.6	25.6
25	23.9	28.2	32.9	37.2	39.4	40.1	39.7	37.9	34.3	29.5	24.9	22.7
30	21.1	25.7	31.2	36.5	39.6	40.8	40.1	37.6	32.7	27.3	22.2	19.7
35	18.1	23.1	29.2	35.6	39.6	41.2	40.4	37.0	31.4	24.9	19.3	16.7
40	15.1	20.3	27.1	34.4	39.3	41.4	40.3	36.2	29.6	22.2	16.4	13.6
45	12.0	17.5	24.7	32.9	38.8	41.4	40.1	35.2	27.6	19.6	13.4	10.6
50	8.18	14.5	22.2	31.2	38.1	41.2	39.7	33.9	25.3	16.7	10.4	7.61
55	6.09	11.5	20.1	29.4	36.5	40.9	39.2	32.5	22.9	13.8	7.41	4.80
60	3.40	8.50	16.6	27.3	36.2	40.6	38.6	30.8	20.3	10.8	4.61	2.28

2.4 THE AVAILABILITY OF SOLAR RADIATION DATA

The most detailed solar radiation data available is in the Monthly Radiation Summary of the Atmospheric Environment Service of Environment Canada. This provides hour-by-hour total horizontal radiation for fifty Canadian sites, and at several of these sites also provides diffuse radiation, reflected solar radiation, net radiation or total illumination.

This same data is also available on magnetic tape for those who are using the computer to do hour-by-hour analysis of a solar energy system. This data can be obtained from the Atmospheric Environment Service, Canadian Climate Center, 4905 Dufferin St., Downsview, Ontario, M3H 5T4, Phone (416) 667-4626.

Where no radiation data is available, it is possible to estimate the clear sky radiation by the method of Section 2.8, D and B. Note also that Tables 18 through 26, beginning on Page 27.19 of the 1981 ASHRAE Handbook of Fundamentals provide the direct normal beam radiation for average clear sky conditions for all latitudes and times. Using clear sky radiation, the methods of Section 2.7 and 2.9, from D and B, may be used to estimate the average solar radiation and to give it a reasonable statistical distribution.

Finally, the Solar Programs Office of Public Works Canada provides the monthly average daily total radiation on a horizontal surface for 97 Canadian cities. This data is presented in Appendix 2A at the end of this chapter.

2.5 BEAM AND DIFFUSE COMPONENTS OF MONTHLY AVERAGE RADIATION

Usually, the solar data available (Appendix 2A, for example) is for total horizontal radiation. The geometric factor, R_b, converts the beam radiation on a horizontal surface to that on a tilted surface, but this does not apply to the diffuse component. Using the nomenclature of Figure 2.5.1, the radiation on a horizontal surface is given by

$$G_h = G_b \text{ Cos } \Theta_z + G_d \qquad \text{(Eqn 2.5.1)}$$

and on a tilted surface by

$$\begin{aligned} G_t = G_b \text{ Cos } \Theta &+ G_d \, (1 + \text{Cos } \beta)/2 \\ &+ G_h \, \rho_g \, (1 - \text{Cos } \beta)/2 \end{aligned} \qquad \text{(Eqn 2.5.2)}$$

$(1+\text{Cos}\beta)/2$ is the view factor from the sky to a tilted collector and $(1-\text{Cos}\beta)/2$ is the view factor from the ground.

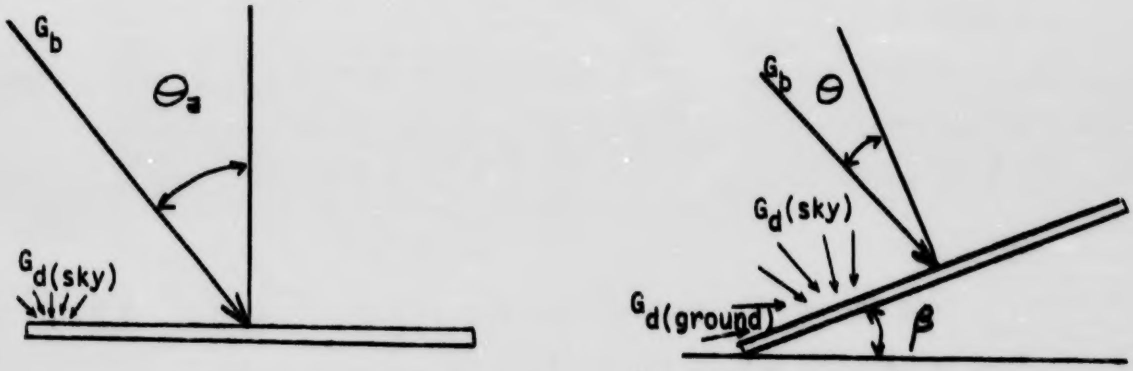


FIGURE 2.5.1

GEOMETRY OF BEAM

AND DIFFUSE RADIATION

The weather data provides G_h, not G_b or G_d, so there is no way of solving these equations and G_d must be estimated from G_h.

There are several methods of making this estimate. One such method is appropriate for dealing with monthly mean daily total radiation, the kind needed as an input to f-Chart. It is based on the use of $\overline{R}$, the ratio of the monthly mean radiation on a tilted surface to the monthly mean radiation on a horizontal surface. This ratio depends on the ratio

$$\overline{K}_T = \overline{H}/\overline{H}_o$$

since $\overline{K}_T$ is a measure of the amount of scattering in the atmosphere, and therefore on the ratio of diffuse to direct radiation. Appendix 2A2 at the end of this chapter contains graphs of $\overline{R}$ as a function of $\overline{K}_T$.

Example 1

Find $\overline{H}_T$ for Winnipeg in January for a collector in a vertical position facing south.

Solution

From the monthly mean weather data of Appendix 2A-1, $\overline{H} = 5$ MJ/m² day.
From Table 2.3.1. $\overline{H}_o = 719$ BTU/ft² day at latitude 50°

$$\overline{H}_o = 719 \times .011357 = 8.17 \text{ MJ/m}^2 \text{ day}$$

$$\therefore \overline{K}_T = \frac{\overline{H}}{\overline{H}_o} = \frac{5}{8.17} = .61$$

From Graph 2A.2 - 27 $\overline{R} = 2.95$

$$\text{Therefore } \overline{H}_T = \overline{R}\ \overline{H}$$

$$= 2.95 \times 5$$

$$= 14.75 \text{ MJ/m}^2 \text{ day}$$

Example 2

Find $\overline{H}_T$ for a collector tilted at 45° in Vancouver in June.

Solution

From the monthly mean weather data $\overline{H} = 22$ MJ/m² day. From Table 2.3.1.

$\overline{H}_o = 3621$ BTU/ft² day

$\therefore \overline{H}_o = 3621 \times .011357$

$= 41.1$ MJ/m² day

and $\overline{K}_T = .53$

Interpolating between Figures 5,6,12 and 13 in Appendix 2A.2,

$\overline{R} = .88$

$\therefore \overline{H}_T = \overline{R}\,\overline{H}$

$= .88 \times 22$

$= 19.36$ MJ/m² day

Assignment

1. Find the mean monthly value of the daily solar radiation on a south facing collector at 43° 41' in Toronto in November.

2. Find $\overline{H}_T$ for Medicine Hat for a south facing collector at 35° in July. $\overline{H} = 25$ MJ/m² day.

APPENDIX 2A-1

MONTHLY MEAN WEATHER DATA FOR CANADIAN CITIES

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 ....................... CANADIAN CITY DATA USED IN F-CHART ...................

                                                 JAN  FEB  MAR  APR  MAY  JUN  JUL  AUG  SEP  OCT  NOV  DEC
  1 ABBOTSFORD        BC   D-D (BASED ON 18 C)    515  393  334 [illegible] [illegible] 59 52  52  110  245  370  463
    (LATITUDE= 49.02)      MEAN AMB. TEMP. (C)      1    4    6    9   12   15   17   17   14   10    6    3
                           SOLAR (MJ/M**2/DAY)      3    6   11   16   21   23   23   19   14    7    4    2

  2 AKLAVIK           NWT  D-D (BASED ON 18 C)   1452 1299 1265  930  591  269  152  255  448  786 1147 1496
    (LATITUDE= 68.22)      MEAN AMB. TEMP. (C)    -28  -27  -22  -12    0    9   13   10    3   -7  -19  -26
                           SOLAR (MJ/M**2/DAY)      0    2    8   16   20   22   19   12    6    2    0    0

  3 ARMSTRONG         ONT  D-D (BASED ON 18 C)   1201 1018  893  556  359  161   75  118  267  447  736 1051
    (LATITUDE= 50.28)      MEAN AMB. TEMP. (C)    -21  -18  -11   -1    6   13   16   15    9    4   -7  -16
                           SOLAR (MJ/M**2/DAY)      5    9   14   18   20   22   22   18   12    7    5    3

  4 BAGOTVILLE        QUE  D-D (BASED ON 18 C)   1046  893  762  485  283  100   45   74  198  382  587  921
    (LATITUDE= 48.33)      MEAN AMB. TEMP. (C)    -16  -14   -7    2    9   15   18   17   12    6   -2  -12
                           SOLAR (MJ/M**2/DAY)      5    9   15   17   19   21   20   17   12    8    4    4

  5 BAIE COMEAU       QUE  D-D (BASED ON 18 C)    974  888  762  553  371  157   87   75  233  409  589  841
    (LATITUDE= 49.23)      MEAN AMB. TEMP. (C)    -14  -13   -7    0    7   13   16   15   10    5   -2  -10
                           SOLAR (MJ/M**2/DAY)      4    8   14   17   18   20   19   16   12    7    4    3

  6 BANFF             ALTA D-D (BASED ON 18 C)    906  693  675  470  327  207  114  145  268  427  653  834
    (LATITUDE= 51.18)      MEAN AMB. TEMP. (C)    -11   -7   -4    2    8   11   15   13    9    4   -4   -8
                           SOLAR (MJ/M**2/DAY)      4    8   13   17   21   23   25   20   15    9    5    3

  7 BLUE RIVER        BC   D-D (BASED ON 18 C)    896  656  603  432  259  129   68   72  223  415  615  793
    (LATITUDE= 52.12)      MEAN AMB. TEMP. (C)    -11   -4   -1    4   10   14   16   15   11    6   -2   -7
                           SOLAR (MJ/M**2/DAY)      4    7   12   17   21   23   23   19   16    7    4    3

  8 BRANDON           MAN  D-D (BASED ON 18 C)   1144  934  680  453  254   86   24   46  195  382  699  989
    (LATITUDE= 49.58)      MEAN AMB. TEMP. (C)    -19  -15   -9    3   10   16   19   18   12    6   -5  -14
                           SOLAR (MJ/M**2/DAY)      5    9   14   18   21   23   24   20   14    9    5    4

  9 CALGARY INT'L     ALTA D-D (BASED ON 18 C)    899  719  691  433  269  150   65   97  221  383  619  793
    (LATITUDE= 51.10)      MEAN AMB. TEMP. (C)    -11   -7   -4    3    9   13   17   15   11    6   -3   -8
                           SOLAR (MJ/M**2/DAY)      4    8   13   17   21   23   25   20   15    9    5    3

 10 CASTLEGAR         BC   D-D (BASED ON 18 C)    681  526  468  307  151   58   17   17  106  323  472  621
    (LATITUDE= 49.33)      MEAN AMB. TEMP. (C)     -4   -1    3    8   13   17   21   20   15    8    2   -2
                           SOLAR (MJ/M**2/DAY)      4    7   12   17   21   23   25   21   15    9    5    3

 11 CHARLO            NB   D-D (BASED ON 18 C)    899  795  695  471  285  107   31   54  172  351  530  791
    (LATITUDE= 48.00)      MEAN AMB. TEMP. (C)    -12  -11   -6    1    7   13   17   16   11    5   -1   -8
                           SOLAR (MJ/M**2/DAY)      5    8   13   16   19   20   20   17   13    8    4    4

 12 CHARLOTTETOWN     PEI  D-D (BASED ON 18 C)    767  714  657  470  292  126   27   37  130  290  441  672
    (LATITUDE= 46.28)      MEAN AMB. TEMP. (C)     -7   -7   -3    2    9   14   18   18   14    9    3   -4
                           SOLAR (MJ/M**2/DAY)      5    9   13   15   18   21   21   18   13    8    5    4

 13 CHATHAM           NB   D-D (BASED ON 18 C)    848  755  663  451  266  101   24   43  148  325  502  758
    (LATITUDE= 47.03)      MEAN AMB. TEMP. (C)     -9   -9   -3    3   10   15   19   18   13    8    1   -7
                           SOLAR (MJ/M**2/DAY)      5    8   13   16   19   20   20   17   14    8    5    4

 14 CHIBOUGAMAU       QUE  D-D (BASED ON 18 C)   1181 1045  909  606  362  121   59  112  294  477  694 1054
    (LATITUDE= 49.50)      MEAN AMB. TEMP. (C)    -20  -18  -10   -1    6   13   16   15   10    4   -4  -15
                           SOLAR (MJ/M**2/DAY)      5    9   15   17   19   21   19   14   12    7    4    3
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.... CANADIAN CITY DATA USED IN F-CHART ....

			JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
15 CHURCHILL	MAN	D-D (BASED ON 18 C)	1414	1261	1187	871	620	359	194	204	366	590	902	1234
(LATITUDE= 58.78)		MEAN AMB. TEMP. (C)	-26	-27	-20	-11	-2	6	12	12	6	-1	-12	-22
		SOLAR (MJ/M**2/DAY)	2	6	13	19	21	22	21	16	9	4	2	1
16 COMOX	BC	D-D (BASED ON 18 C)	494	398	404	302	185	93	43	51	129	276	380	448
(LATITUDE= 49.72)		MEAN AMB. TEMP. (C)	2	4	5	8	12	15	17	17	14	9	5	4
		SOLAR (MJ/M**2/DAY)	3	6	10	16	20	22	23	18	13	7	4	2
17 CORONATION	ALTA	D-D (BASED ON 18 C)	1061	854	779	463	247	124	49	80	221	410	696	922
(LATITUDE= 52.10)		MEAN AMB. TEMP. (C)	-16	-12	-7	3	10	14	17	16	11	4	-5	-12
		SOLAR (MJ/M**2/DAY)	4	8	14	18	21	23	24	19	14	9	5	3
18 CRANBROOK	BC	D-D (BASED ON 18 C)	829	641	577	367	227	118	41	61	185	378	590	748
(LATITUDE= 49.50)		MEAN AMB. TEMP. (C)	-9	-5	-1	6	11	14	18	17	12	5	-2	-10
		SOLAR (MJ/M**2/DAY)	4	8	13	17	21	23	25	21	15	9	5	3
19 CREE LAKE	SASK	D-D (BASED ON 18 C)	1351	1092	987	581	327	128	83	117	333	536	877	1215
(LATITUDE= 57.30)		MEAN AMB. TEMP. (C)	-24	-20	-12	-2	6	13	16	15	8	2	-11	-20
		SOLAR (MJ/M**2/DAY)	3	6	10	19	23	23	22	17	10	6	3	2
20 DAUPHIN	MAN	D-D (BASED ON 18 C)	1159	950	849	485	261	98	32	54	198	370	693	1001
(LATITUDE= 51.15)		MEAN AMB. TEMP. (C)	-19	-16	-9	2	10	15	19	17	12	6	-5	-14
		SOLAR (MJ/M**2/DAY)	5	9	14	18	21	22	23	19	13	8	4	4
21 DAWSON	YUK	D-D (BASED ON 18 C)	1443	1152	996	593	318	127	81	166	347	659	1039	1352
(LATITUDE= 64.07)		MEAN AMB. TEMP. (C)	-29	-23	-14	-2	8	14	16	13	6	-3	-17	-25
		SOLAR (MJ/M**2/DAY)	1	3	9	15	19	21	19	14	7	3	1	0
22 EARLTON	ONT	D-D (BASED ON 18 C)	1058	897	792	483	274	106	52	94	204	375	607	934
(LATITUDE= 47.70)		MEAN AMB. TEMP. (C)	-16	-14	-8	2	9	15	18	16	11	6	-2	-12
		SOLAR (MJ/M**2/DAY)	5	9	15	17	20	22	21	18	12	8	4	4
23 EDMONTON	ALTA	D-D (BASED ON 18 C)	1016	806	727	421	222	109	47	62	215	398	666	890
(LATITUDE= 53.57)		MEAN AMB. TEMP. (C)	-15	-11	-5	4	11	15	19	16	11	5	-4	-11
		SOLAR (MJ/M**2/DAY)	4	7	13	18	20	23	23	18	13	8	4	3
24 ESTEVAN	SASK	D-D (BASED ON 18 C)	1050	854	750	423	228	84	20	37	177	355	652	911
(LATITUDE= 49.20)		MEAN AMB. TEMP. (C)	-16	-13	-6	4	11	16	20	19	13	7	-4	-12
		SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	25	20	15	10	5	4
25 FORT MCMURRAY	ALTA	D-D (BASED ON 18 C)	1237	978	850	497	279	141	69	112	271	463	795	1086
(LATITUDE= 56.65)		MEAN AMB. TEMP. (C)	-22	-17	-9	1	9	14	16	15	9	3	-8	-17
		SOLAR (MJ/M**2/DAY)	3	6	12	19	21	23	22	18	11	6	3	2
26 FORT NELSON	BC	D-D (BASED ON 18 C)	1276	994	843	505	259	112	58	107	275	523	910	1201
(LATITUDE= 58.83)		MEAN AMB. TEMP. (C)	-23	-17	-9	1	10	15	17	15	9	1	-12	-21
		SOLAR (MJ/M**2/DAY)	2	5	10	16	20	20	20	16	10	5	2	1
27 FORT SMITH	NWT	D-D (BASED ON 18 C)	1403	1145	1010	637	336	153	79	124	310	542	897	1216
(LATITUDE= 60.00)		MEAN AMB. TEMP. (C)	-27	-23	-15	-3	7	13	16	14	8	1	-12	-21
		SOLAR (MJ/M**2/DAY)	2	5	10	17	21	21	21	16	10	4	2	1
28 FORT ST. JOHN	BC	D-D (BASED ON 18 C)	1093	846	758	465	254	136	77	112	242	430	736	970
(LATITUDE= 56.25)		MEAN AMB. TEMP. (C)	-17	-12	-6	3	10	14.	16	15	10	4	-7	-13
		SOLAR (MJ/M**2/DAY)	3	6	12	16	20	23	21	17	11	6	3	2

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[illegible]

                                                      JAN  FEB  MAR  APR  MAY  JUN  JUL  AUG  SEP  OCT  NOV  DEC
29  FREDERICTON      NB   D-D (BASED ON 18 C)  [illegible] 727  615  418  225   68   21   35  147  315  499  [illegible]
    (LATITUDE= 45.87)     MEAN AMB. TEMP. (C)        -9   -6   -2    4   11   16   19   18   13    8    2   -6
                          SOLAR (MJ/M**2/DAY)         5    8   13   15   18   21   20   17   13    9    5    4

30  FROBISHER BAY    NWT  D-D (BASED ON 18 C)      1369 1236 1254  962  660  434  312  342  467  706  912 1191
    (LATITUDE= 63.75)     MEAN AMB. TEMP. (C)       -26  -25  -22  -14   -3    4    8    7    2   -5  -12  -20
                          SOLAR (MJ/M**2/DAY)         1    4   10   17   23   21   16   13    8    3    1    0

31  GANDER INT'L     NFLD D-D (BASED ON 18 C)       747  687  669  516  362  202   73   84  189  362  483  665
    (LATITUDE= 48.95)     MEAN AMB. TEMP. (C)        -6   -6   -4    1    6   11   17   16   12    6    2   -3
                          SOLAR (MJ/M**2/DAY)         4    7   11   14   17   18   18   16   12    7    4    3

32  GOOSE BAY        NFLD D-D (BASED ON 18 C)      1065  914  821  594  407  214   91  118  248  458  650  942
    (LATITUDE= 53.32)     MEAN AMB. TEMP. (C)       -16  -14   -8   -2    5   11   16   15   10    3   -4  -12
                          SOLAR (MJ/M**2/DAY)         3    7   12   16   18   18   18   14   10    6    3    3

33  GRANDE PRAIRIE   ALTA D-D (BASED ON 18 C)      1095  861  790  469  249  133   75  107  236  431  731  969
    (LATITUDE= 55.17)     MEAN AMB. TEMP. (C)       -17  -12   -7    2   10   14   16   15   10    4   -6  -13
                          SOLAR (MJ/M**2/DAY)         4    6   10   16   20   27   21   21   13    7    3    2

34  HALIFAX          NS   D-D (BASED ON 18 C)       676  628  582  419  278  133   37   30  105  251  391  593
    (LATITUDE= 44.65)     MEAN AMB. TEMP. (C)        -3   -3    0    5   10   14   18   19   16   11    6   -1
                          SOLAR (MJ/M**2/DAY)         5    8   12   15   17   20   19   18   14    9    5    4

35  INUVIK           NWT  D-D (BASED ON 18 C)      1506 1337 1320  966  594  244  147  231  458  801 1189 1391
    (LATITUDE= 68.30)     MEAN AMB. TEMP. (C)       -29  -29  -24  -15   -1   10   13   10    3   -7  -21  -27
                          SOLAR (MJ/M**2/DAY)         0    2    8   16   20   22   18   12    6    2    1    0

36  JASPER           ALTA D-D (BASED ON 18 C)       936  694  643  440  288  166   96  126  242  411  654  836
    (LATITUDE= 52.88)     MEAN AMB. TEMP. (C)       -12   -7   -3    3    9   13   15   14   10    5   -4   -9
                          SOLAR (MJ/M**2/DAY)         3    7   12   17   21   23   23   19   15    7    4    3

37  KAMLOOPS         BC   D-D (BASED ON 18 C)       742  543  458  272  128   41   11   20  106  298  495  642
    (LATITUDE= 50.68)     MEAN AMB. TEMP. (C)        -6   -1    4    9   14   18   21   20   15    8    2   -3
                          SOLAR (MJ/M**2/DAY)         3    6   12   17   21   23   23   19   14    7    4    3

38  KAPUSKASING      ONT  D-D (BASED ON 18 C)      1123  956  841  521  319  136   70  105  232  401  667  995
    (LATITUDE= 49.42)     MEAN AMB. TEMP. (C)       -18  -16   -9    1    8   14   17   15   10    5   -4  -14
                          SOLAR (MJ/M**2/DAY)         5    8   13   16   18   22   20   17   12    7    4    3

39  KENORA           ONT  D-D (BASED ON 18 C)      1109  924  776  465  259   91   27   47  196  373  678  996
    (LATITUDE= 49.80)     MEAN AMB. TEMP. (C)       -18  -15   -7    3   10   16   19   18   12    6   -5  -14
                          SOLAR (MJ/M**2/DAY)         5    9   14   18   20   23   23   19   13    8    5    4

40  KINGSTON         ONT  D-D (BASED ON 18 C)       793  702  612  377  202   57    9   17   87  256  439  711
    (LATITUDE= 44.25)     MEAN AMB. TEMP. (C)        -7   -8   -2    5   12   17   21   20   16   10    4   -3
                          SOLAR (MJ/M**2/DAY)         5    8   13   16   20   22   22   19   15    9    5    4

41  LETHBRIDGE       ALTA D-D (BASED ON 18 C)       851  666  634  383  221  106   29   48  168  323  560  729
    (LATITUDE= 49.70)     MEAN AMB. TEMP. (C)        -9   -6   -2    5   11   15   19   18   13    8   -1   -6
                          SOLAR (MJ/M**2/DAY)         5    8   14   18   22   24   25   21   15   10    6    4

42  LONDON           ONT  D-D (BASED ON 18 C)       746  668  582  342  183   48   12   22   98  251  446  670
    (LATITUDE= 42.98)     MEAN AMB. TEMP. (C)        -6   -6   -1    7   12   18   21   20   16   10    3   -4
                          SOLAR (MJ/M**2/DAY)         5    8   13   16   21   23   23   19   15   10    5    5
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.... CANADIAN CITY DATA USED IN [illegible] ....

				JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
43	MANIWAKI	QUE	D-D (BASED ON 18 C)	975	841	722	429	243	57	40	67	176	346	539	869
	(LATITUDE= 46.38)		MEAN AMB. TEMP. (C)	-13	-12	-5	4	10	16	18	17	12	7	0	-10
			SOLAR (MJ/M**2/DAY)	5	9	14	16	20	22	21	18	13	9	5	4
44	MEDICINE HAT	ALTA	D-D (BASED ON 18 C)	934	739	659	361	189	79	17	34	157	324	589	793
	(LATITUDE= 50.02)		MEAN AMB. TEMP. (C)	-12	-8	-3	6	12	16	20	19	13	8	-2	-8
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	25	21	15	10	5	4
45	MONCTON	NB	D-D (BASED ON 18 C)	806	722	647	447	261	109	26	44	145	308	472	722
	(LATITUDE= 46.10)		MEAN AMB. TEMP. (C)	-8	-8	-3	4	10	15	19	18	13	8	2	-5
			SOLAR (MJ/M**2/DAY)	4	8	12	16	18	19	20	17	13	9	4	4
46	MONT JOLI	QUE	D-D (BASED ON 18 C)	900	798	713	487	305	129	57	77	191	363	538	795
	(LATITUDE= 48.60)		MEAN AMB. TEMP. (C)	-11	-10	-5	2	8	14	17	16	12	6	0	-8
			SOLAR (MJ/M**2/DAY)	5	8	14	17	19	20	20	17	12	8	4	4
47	MONTREAL INT'L	QUE	D-D (BASED ON 18 C)	865	757	636	369	168	38	7	20	106	269	471	765
	(LATITUDE= 45.50)		MEAN AMB. TEMP. (C)	-10	-9	-2	6	13	19	21	20	15	9	2	-7
			SOLAR (MJ/M**2/DAY)	5	9	13	16	19	21	21	18	14	9	5	4
48	MOOSONEE	ONT	D-D (BASED ON 18 C)	1163	1020	928	608	392	200	106	130	244	414	674	1032
	(LATITUDE= 51.27)		MEAN AMB. TEMP. (C)	-20	-18	-12	-2	5	12	16	14	10	5	-4	-15
			SOLAR (MJ/M**2/DAY)	4	8	13	17	18	20	18	15	11	6	3	3
49	MOUNT FOREST	ONT	D-D (BASED ON 18 C)	828	755	654	406	245	91	41	61	140	299	490	745
	(LATITUDE= 43.97)		MEAN AMB. TEMP. (C)	-9	-9	-3	5	10	16	18	17	14	8	2	-6
			SOLAR (MJ/M**2/DAY)	5	8	12	16	20	23	23	19	15	10	5	4
50	MUSKOKA	ONT	D-D (BASED ON 18 C)	869	762	674	401	220	80	33	51	150	306	505	775
	(LATITUDE= 44.97)		MEAN AMB. TEMP. (C)	-10	-9	-4	5	11	16	19	18	14	8	1	-7
			SOLAR (MJ/M**2/DAY)	5	8	13	16	21	23	23	19	14	9	5	4
51	NATASHQUAN	QUE	D-D (BASED ON 18 C)	927	823	746	560	414	239	125	145	259	419	563	832
	(LATITUDE= 50.20)		MEAN AMB. TEMP. (C)	-12	-11	-6	-1	5	10	14	13	9	5	-1	-9
			SOLAR (MJ/M**2/DAY)	4	7	12	16	18	19	19	16	12	7	4	3
52	NORTH BAY	ONT	D-D (BASED ON 18 C)	956	823	725	444	248	87	37	62	176	343	564	853
	(LATITUDE= 46.32)		MEAN AMB. TEMP. (C)	-13	-11	-5	3	10	16	18	17	12	7	-1	-9
			SOLAR (MJ/M**2/DAY)	5	9	14	17	20	23	22	19	13	9	5	4
53	OTTAWA INT'L	ONT	D-D (BASED ON 18 C)	899	778	654	371	181	44	10	28	123	290	499	796
	(LATITUDE= 45.32)		MEAN AMB. TEMP. (C)	-11	-10	-3	6	12	18	21	19	15	9	1	-8
			SOLAR (MJ/M**2/DAY)	6	9	14	15	20	22	22	18	14	9	5	4
54	PEACE RIVER	ALTA	D-D (BASED ON 18 C)	1163	881	803	468	267	130	71	111	263	447	800	1020
	(LATITUDE= 56.25)		MEAN AMB. TEMP. (C)	-21	-14	-9	2	9	14	16	14	9	4	-9	-15
			SOLAR (MJ/M**2/DAY)	3	6	12	18	20	20	22	18	12	7	3	2
55	PENTICTON	BC	D-D (BASED ON 18 C)	649	499	445	278	147	52	13	19	107	287	448	570
	(LATITUDE= 49.47)		MEAN AMB. TEMP. (C)	-3	0	4	9	13	17	20	19	15	9	3	0
			SOLAR (MJ/M**2/DAY)	3	7	12	17	21	23	24	20	15	8	4	3
56	PORT HARDY	BC	D-D (BASED ON 18 C)	494	401	421	345	262	186	136	131	183	289	380	451
	(LATITUDE= 50.67)		MEAN AMB. TEMP. (C)	2	4	4	7	10	12	14	14	12	9	5	3
			SOLAR (MJ/M**2/DAY)	3	6	9	13	18	19	20	15	11	6	3	2

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....................... CANADIAN CITY DATA USED IN F-CHART . ....................

                                                          JAN   FEB   MAR   APR   MAY   JUN   JUL   AUG   SEP   OCT   NOV   DEC
57  POSTE DE LA BA    QUE   D-D (BASED ON 18 C)          1224  1145  1056   752   521   369   200   251   317   491   681  1037
    (LATITUDE= 55.28)       MEAN AMB. TEMP. (C)           -22   -22   -17    -7     1     6    11    12     8     2    -5   -15
                            SOLAR (MJ/M**2/DAY)             3     7    12    17    18    20    18    14     9     5     3     2

58  PRINCE ALBERT     SASK  D-D (BASED ON 18 C)          1204   987   883   489   265   119    45    74   234   436   752  1073
    (LATITUDE= 53.17)       MEAN AMB. TEMP. (C)           -21   -17   -10     2    10    14    18    16    10     4    -7   -16
                            SOLAR (MJ/M**2/DAY)             4     8    14    19    21    22    23    19    13     8     4     3

59  PRINCE GEORGE     BC    D-D (BASED ON 18 C)           925   685   623   422   265   152   102   135   245   413   626   795
    (LATITUDE= 53.83)       MEAN AMB. TEMP. (C)           -12    -6    -2     4     9    13    15    14    10     5    -3    -8
                            SOLAR (MJ/M**2/DAY)             3     6    11    16    19    22    21    18    12     6     3     2

60  PRINCE RUPERT     BC    D-D (BASED ON 18 C)           578   411   455   390   309   213   168   157   193   303   418   522
    (LATITUDE= 54.28)       MEAN AMB. TEMP. (C)             0     3     3     5     9    11    13    13    12     8     4     1
                            SOLAR (MJ/M**2/DAY)             3     5    10    14    18    19    17    14     9     5     3     1

61  QUEBEC            QUE   D-D (BASED ON 18 C)           919   810   692   441   232    76    22    45   155   332   533   823
    (LATITUDE= 46.81)       MEAN AMB. TEMP. (C)           -12   -11    -4     3    11    16    19    18    13     7     0    -9
                            SOLAR (MJ/M**2/DAY)             5     9    14    17    19    21    21    17    13     8     5     4

62  RED DEER          ALTA  D-D (BASED ON 18 C)          1021   817   751   443   250   143    69   106   228   416   679   902
    (LATITUDE= 52.27)       MEAN AMB. TEMP. (C)           -15   -11    -6     3    10    13    16    15    10     5    -5   -11
                            SOLAR (MJ/M**2/DAY)             4     8    13    18    21    23    22    17    14     9     5     3

63  REGINA            SASK  D-D (BASED ON 18 C)          1094   914   817   441   234    97    29    46   198   394   697   959
    (LATITUDE= 50.45)       MEAN AMB. TEMP. (C)           -17   -14    -8     3    11    15    19    18    12     5    -5   -13
                            SOLAR (MJ/M**2/DAY)             5     9    14    18    21    23    24    20    14     9     5     4

64  RESOLUTE          NWT   D-D (BASED ON 18 C)          1567  1456  1527  1232   889   549   425   474   687  1016  1275  1452
    (LATITUDE= 74.68)       MEAN AMB. TEMP. (C)           -33   -34   -31   -23   -11     0     4     3    -5   -15   -24   -29
                            SOLAR (MJ/M**2/DAY)             0     1     5    15    23    25    18    11     5     1     0     0

65  REVELSTOKE        BC    D-D (BASED ON 18 C)           742   561   500   323   163    68    21    36   130   344   516   669
    (LATITUDE= 51.00)       MEAN AMB. TEMP. (C)            -7    -3     1     6    13    16    19    17    13     7     1    -4
                            SOLAR (MJ/M**2/DAY)             4     7    12    17    21    23    24    20    14     8     4     4

66  RIVIERE AU REN    QUE   D-D (BASED ON 18 C)           849   788   717   523   351   160    77    92   201   371   520   738
    (LATITUDE= 49.02)       MEAN AMB. TEMP. (C)           -10    -9    -5     0     6    12    16    16    12     6     1    -6
                            SOLAR (MJ/M**2/DAY)             4     5    10    16    18    17    19    16    12     7     4     3

67  RIVIER DU LOUP    QUE   D-D (BASED ON 18 C)           942   833   728   507   305   108    42    83   202   386   568   833
    (LATITUDE= 47.80)       MEAN AMB. TEMP. (C)           -12   -12    -6     1     8    14    17    16    12     6    -1    -9
                            SOLAR (MJ/M**2/DAY)             5     9    14    17    19    21    20    17    13     8     4     4

68  SAINT JOHN        NB    D-D (BASED ON 18 C)           790   721   640   447   286   135    49    62   162   313   464   702
    (LATITUDE= 45.27)       MEAN AMB. TEMP. (C)            -7    -7    -1     4    10    14    17    17    14     9     3    -4
                            SOLAR (MJ/M**2/DAY)             5     9    13    16    19    21    20    18    14     9     5     4

69  SASKATOON         SASK  D-D (BASED ON 18 C)          1139   936   829   445   233    97    30    54   204   402   714   994
    (LATITUDE= 52.23)       MEAN AMB. TEMP. (C)           -19   -15    -9     3    11    15    19    17    11     5    -6   -14
                            SOLAR (MJ/M**2/DAY)             4     8    13    17    21    22    24    19    14     8     4     3

70  SCHEFFERVILLE     QUE   D-D (BASED ON 18 C)          1261  1106  1016   750   529   287   172   227   372   598   803  1118
    (LATITUDE= 54.80)       MEAN AMB. TEMP. (C)           -23   -21   -15    -7     1     9    13    11     6    -1    -9   -18
                            SOLAR (MJ/M**2/DAY)             3     7    13    17    18    19    17    15    10     6     3     2
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                                                   JAN   FEB   MAR   APR   MAY   JUN   JUL   AUG   SEP   OCT   NOV   DEC
71  SEPT-ILLES          QUE  D-D (BASED ON 18 C)   [illegible]   867   767   549   [illegible]   194    95   122   252   436   607   875
    (LATITUDE= 50.22)        MEAN AMB. TEMP. (C)   -14   -13    -7     0     6    12    15    14    10     4    -2   -10
                             SOLAR (MJ/M**2/DAY)     5     9    12    14    19    20    18    16    12     8     4     3

72  SHERBROOKE          QUE  D-D (BASED ON 18 C)   918   825   697   438   249    91    49    76   188   337   536   828
    (LATITUDE= 45.40)        MEAN AMB. TEMP. (C)   -12   -11    -4     4    10    15    18    17    12     7     1    -8
                             SOLAR (MJ/M**2/DAY)     5     9    13    16    19    22    21    18    14     9     5     4

73  SIMCO               ONT  D-D (BASED ON 18 C)   731   662   567   343   176    44    12    22    98   246   414   647
    (LATITUDE= 42.85)        MEAN AMB. TEMP. (C)    -6    -6     0     7    12    18    20    19    16    10     4    -3
                             SOLAR (MJ/M**2/DAY)     5     8    13    16    20    23    22    19    15    10     5     4

74  ST. JOHN'S         NFLD  D-D (BASED ON 18 C)   674   629   634   507   387   229    98    91   184   337   425   599
    (LATITUDE= 47.57)        MEAN AMB. TEMP. (C)    -4    -4    -2     1     6    10    15    15    12     7     4    -1
                             SOLAR (MJ/M**2/DAY)     3     6    10    13    17    18    18    14    12     7     3     3

75. STEPHENVILLE       NFLD  D-D (BASED ON 18 C)   707   674   646   490   346   197    75    67   172   328   447   634
    (LATITUDE= 48.53)        MEAN AMB. TEMP. (C)    -5    -6    -3     2     7    12    16    16    12     7     3    -2
                             SOLAR (MJ/M**2/DAY)     4     7    12    14    18    19    19    16    12     8     4     3

76  SUDBURY             ONT  D-D (BASED ON 18 C)   982   860   752   452   255    87    33    59   176   348   572   871
    (LATITUDE= 46.48)        MEAN AMB. TEMP. (C)   -13   -12    -6     3    10    16    19    18    13     7    -1   -10
                             SOLAR (MJ/M**2/DAY)     5     9    14    17    21    23    23    19    13     8     4     4

77  SWIFT CURRENT      SASK  D-D (BASED ON 18 C)   991   809   742   429   242   110    33    54   194   370   648   920
    (LATITUDE= 50.33)        MEAN AMB. TEMP. (C)   -14   -11    -6     4    10    15    19    18    12     6    -4   -10
                             SOLAR (MJ/M**2/DAY)     5     9    14    18    21    23    25    20    14    10     5     4

78  SYDNEY               NS  D-D (BASED ON 18 C)   696   664   633   475   320   158    41    41   130   293   413   605
    (LATITUDE= 46.17)        MEAN AMB. TEMP. (C)    -4    -6    -2     2     8    13    18    18    14     9     4    -2
                             SOLAR (MJ/M**2/DAY)     4     8    12    14    19    20    20    17    13     8     4     3

79  TERRACE              BC  D-D (BASED ON 18 C)   734   536   504   369   244   130    77    90   184   365   534   663
    (LATITUDE= 54.52)        MEAN AMB. TEMP. (C)    -6    -1     2     6    10    14    16    16    12     6     0    -3
                             SOLAR (MJ/M**2/DAY)     2     5    10    14    18    19    18    15     9     5     3     2

80  THE PAS             MAN  D-D (BASED ON 18 C)  1253  1026   913   554   322   130    40    74   237   439   766  1098
    (LATITUDE= 53.82)        MEAN AMB. TEMP. (C)   -22   -18   -11     0     8    14    18    16    10     4    -8   -17
                             SOLAR (MJ/M**2/DAY)     3     7    13    18    20    20    21    17    11     6     3     3

81  THOMPSON            MAN  D-D (BASED ON 18 C)  1381  1136   993   619   388   178    97   137   329   573   874  1225
    (LATITUDE= 55.80)        MEAN AMB. TEMP. (C)   -26   -22   -14    -3     6    12    15    14     7     0   -11   -22.
                             SOLAR (MJ/M**2/DAY)     3     7    13    19    22    22    22    16    10     6     3     2

82  THUNDER BAY         ONT  D-D (BASED ON 18 C)  1017   877   749   469   301   131    47    71   204   371   616   893
    (LATITUDE= 48.37)        MEAN AMB. TEMP. (C)   -15   -13    -6     2     8    14    18    17    11     6    -3   -11
                             SOLAR (MJ/M**2/DAY)     5     9    14    18    20    23    23    19    13     8     5     4

83  TIMMINS             ONT  D-D (BASED ON 18 C)  1090   944   824   502   299   124    63    98   225   396   647   977
    (LATITUDE= 48.47)        MEAN AMB. TEMP. (C)   -17   -15    -9     1     9    15    17    16    11     5    -4   -13
                             SOLAR (MJ/M**2/DAY)     5     9    14    17    19    22    21    18    12     7     4     3

84  TORONTO INT'L       ONT  D-D (BASED ON 18 C)   755   671   587   348   187    48    10    19    98   254   438   667
    (LATITUDE= 43.68)        MEAN AMB. TEMP. (C)    -6    -6    -1     6    12    18    21    20    16    10     3    -4
                             SOLAR (MJ/M**2/DAY)     5     8    13    16    20    23    22    19    15    10     5     4
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.... CANADIAN CITY DATA USED IN F-CHART ... .

			JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
85	TRENTON	ONT	D-D (BASED ON 18 C)	734	661	585	345	177	42	7	17	97	255	437	647
	(LATITUDE= 44.10)		MEAN AMB. TEMP. (C)	-7	-6	-1	7	12	16	21	20	16	10	3	-4
			SOLAR (MJ/M**2/DAY)	5	8	13	16	16	23	22	19	15	9	5	4
86	TRURO	NS	D-D (BASED ON 18 C)	772	726	640	469	302	130	40	56	161	303	439	666
	(LATITUDE= 45.43)		MEAN AMB. TEMP. (C)	-6	-6	-2	3	9	13	18	17	13	8	3	-3
			SOLAR (MJ/M**2/DAY)	5	8	12	15	19	21	21	18	14	9	5	4
87	VAL D'OR	QUE	D-D (BASED ON 18 C)	1079	931	827	506	3040	121	66	97	219	399	633	964
	(LATITUDE= 48.05)		MEAN AMB. TEMP. (C)	-17	-14	-8	1	8	15	17	15	11	5	-3	-13
			SOLAR (MJ/M**2/DAY)	5	9	15	17	19	22	21	17	12	8	4	4
88	VANCOUVER INT'L	BC	D-D (BASED ON 18 C)	484	383	379	272	174	85	35	40	114	245	358	440
	(LATITUDE= 49.28)		MEAN AMB. TEMP. (C)	2	4	6	9	12	15	17	17	14	10	6	4
			SOLAR (MJ/M**2/DAY)	3	6	10	15	21	22	23	19	14	7	4	2
89	VICTORIA INT'L	BC	D-D (BASED ON 18 C)	467	375	378	282	191	108	61	65	122	247	353	427
	(LATITUDE= 48.65)		MEAN AMB. TEMP. (C)	3	5	6	9	12	15	16	16	14	10	6	4
			SOLAR (MJ/M**2/DAY)	3	6	11	17	21	23	23	19	15	7	4	3
90	WHITEHORSE	YUK	D-D (BASED ON 18 C)	1143	980	738	543	338	171	125	176	307	537	812	1048
	(LATITUDE= 60.75)		MEAN AMB. TEMP. (C)	-19	-13	-6	0	7	12	14	12	8	1	-9	-16
			SOLAR (MJ/M**2/DAY)	1	4	10	15	20	21	19	14	9	4	2	1
91	WIARTON	ONT	D-D (BASED ON 18 C)	765	702	654	392	243	89	29	35	122	264	451	666
	(LATITUDE= 44.75)		MEAN AMB. TEMP. (C)	-7	-7	-3	5	10	16	19	18	15	10	3	-3
			SOLAR (MJ/M**2/DAY)	5	8	12	16	21	23	23	19	14	9	5	4
92	WILLIAMS LAKE	BC	D-D (BASED ON 18 C)	859	592	567	416	283	138	89	102	228	393	611	798
	(LATITUDE= 52.13)		MEAN AMB. TEMP. (C)	-10	-5	-4	4	10	13	16	15	11	5	-2	-7
			SOLAR (MJ/M**2/DAY)	3	6	11	16	20	22	22	18	13	7	4	2
93	WINDSOR	ONT	D-D (BASED ON 18 C)	691	605	520	295	141	29	3	8	64	204	411	618
	(LATITUDE= 42.33)		MEAN AMB. TEMP. (C)	-4	-3	1	8	14	20	22	21	17	12	4	-2
			SOLAR (MJ/M**2/DAY)	5	8	12	16	20	23	23	19	16	11	5	5
94	WINNIPEG INT'L	MAN	D-D (BASED ON 18 C)	1126	953	810	44	226	76	21	38	175	356	674	993
	(LATITUDE= 49.90)		MEAN AMB. TEMP. (C)	-18	-16	-8	3	11	17	20	19	13	7	-4	-14
			SOLAR (MJ/M**2/DAY)	5	9	14	17	21	22	23	19	13	8	5	4
95	YARMOUTH	NS	D-D (BASED ON 18 C)	641	592	551	401	270	141	55	126	252	252	376	564
	(LATITUDE= 43.83)		MEAN AMB. TEMP. (C)	-3	-3	0	5	9	13	16	16	14	10	5	0
			SOLAR (MJ/M**2/DAY)	5	9	13	15	19	21	21	19	15	9	5	5
96	YELLOWKNIFE	NWT	D-D (BASED ON 18 C)	1445	1235	1138	777	435	178	76	126	333	596	967	1287
	(LATITUDE= 62.47)		MEAN AMB. TEMP. (C)	-29	-26	-19	-8	4	12	16	14	7	-1	-14	-24
			SOLAR (MJ/M**2/DAY)	1	4	10	18	24	23	20	16	8	4	1	1
97	YORKTON	SASK	D-D (BASED ON 18 C)	1163	952	853	480	256	102	34	58	211	400	722	1008
	(LATITUDE= 51.18)		MEAN AMB. TEMP. (C)	-19	-16	-10	2	10	15	19	17	11	5	-6	-14
			SOLAR (MJ/M**2/DAY)	5	9	14	18	21	23	24	19	13	9	5	3

APPENDIX 2A-2

$\overline{R}$ AS A FUNCTION OF $\overline{K}_T$

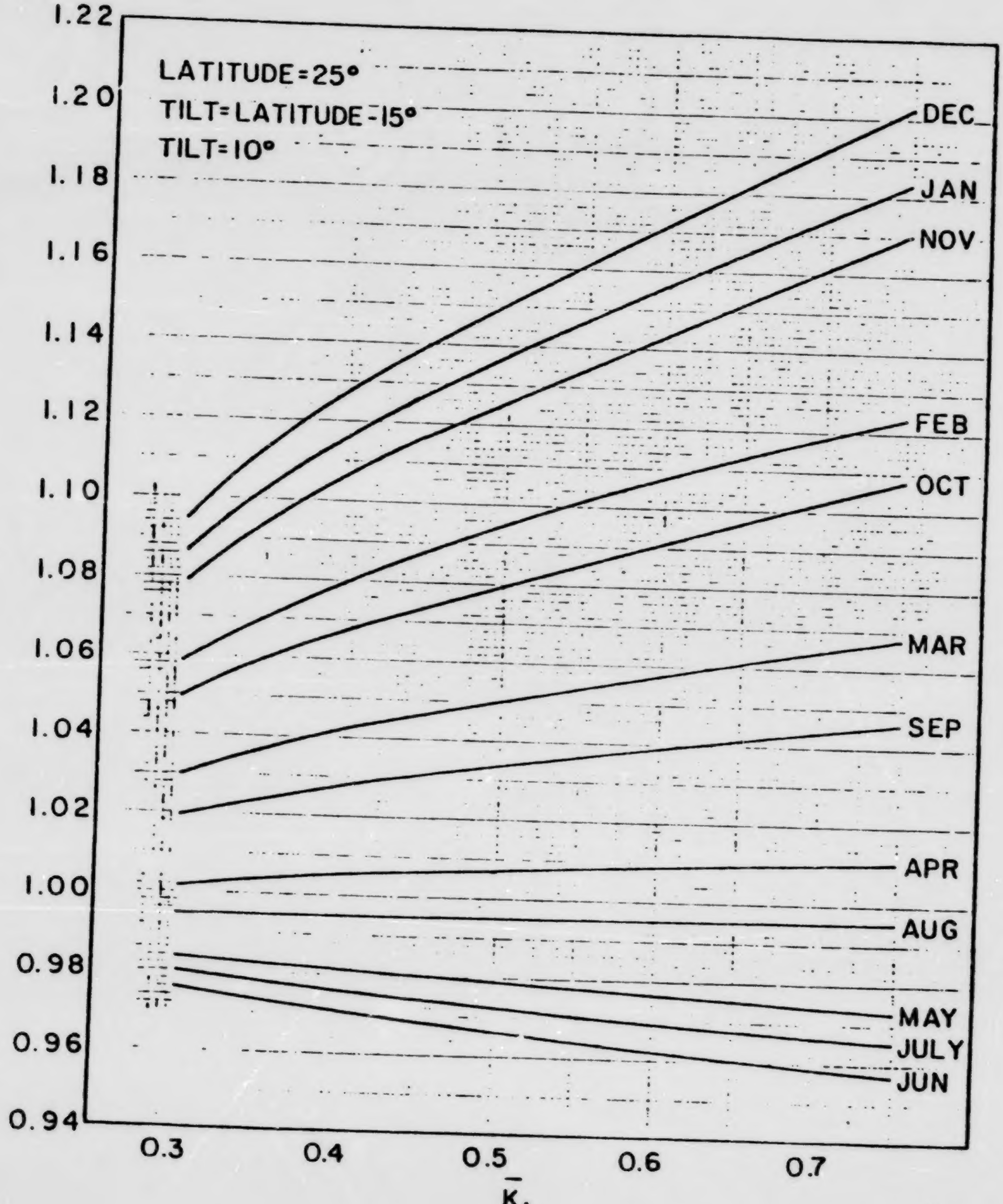

Figure 2A-2-1. Tilt Correction Factor for 25° Latitude, 10° Tilt

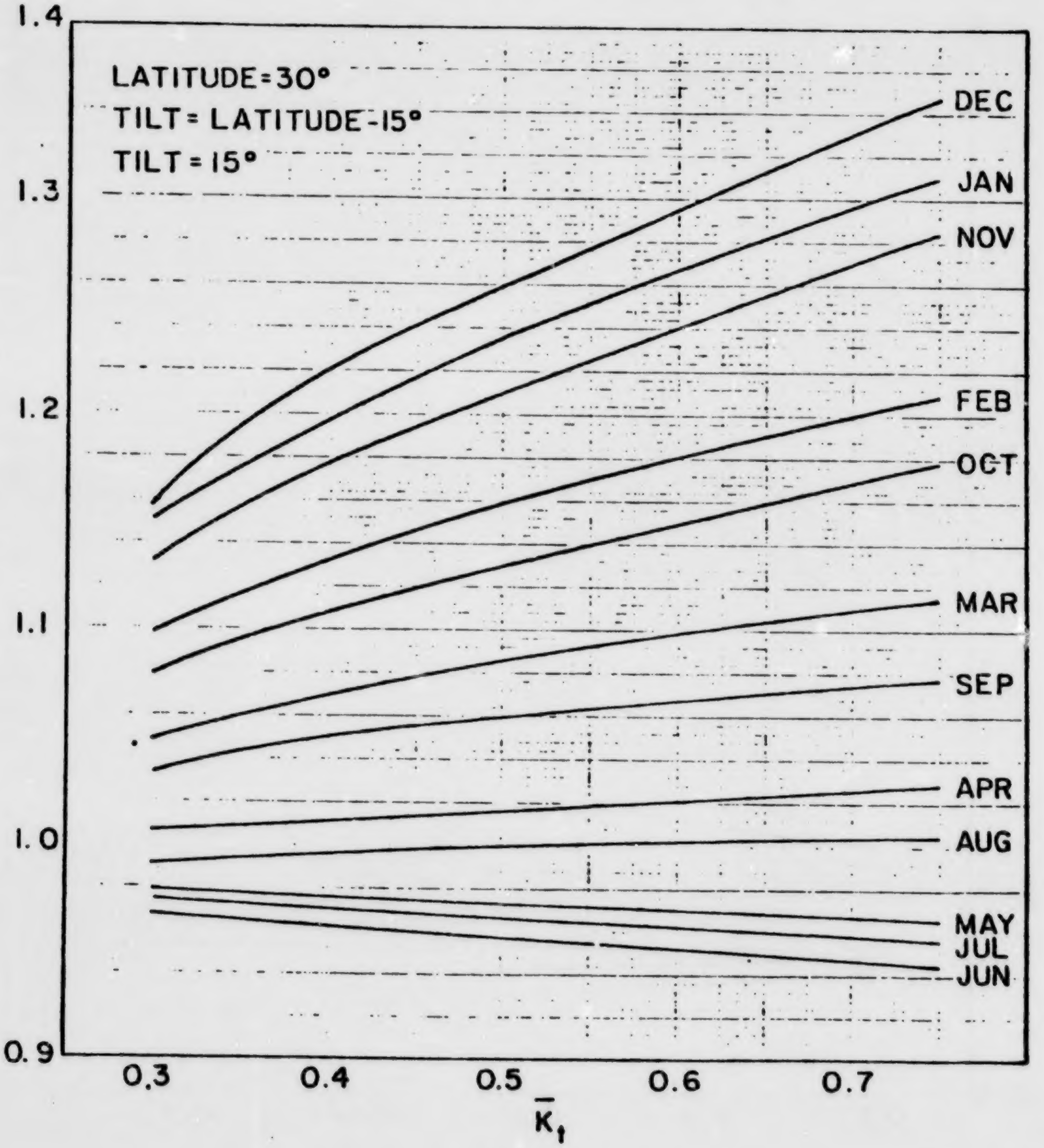

Figure 2A-2-2. Tilt Correction Factor for 30° Latitude, 15° Tilt

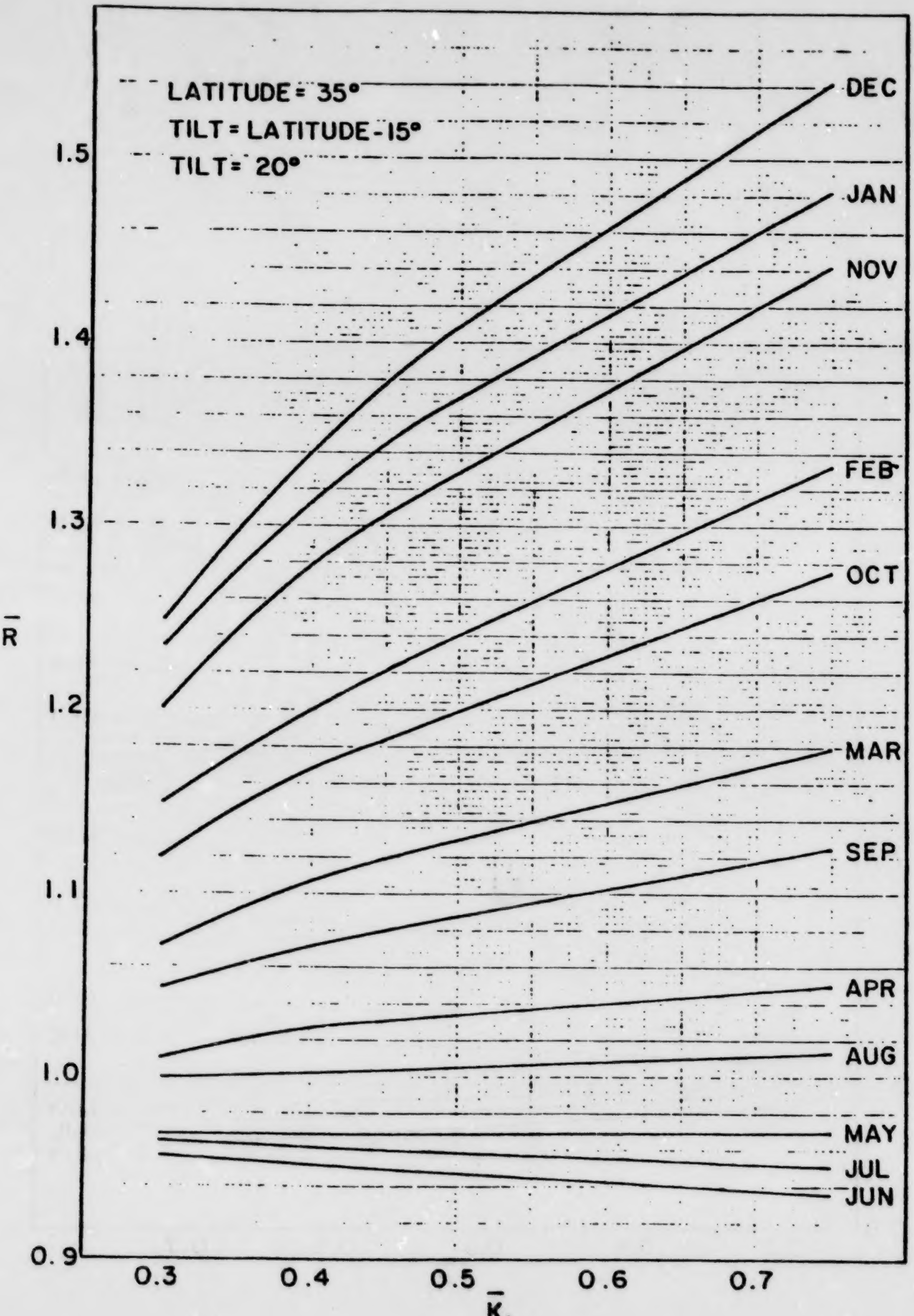

Figure 2A-2-3. Tilt Correction Factor for 35° Latitude, 20° Tilt

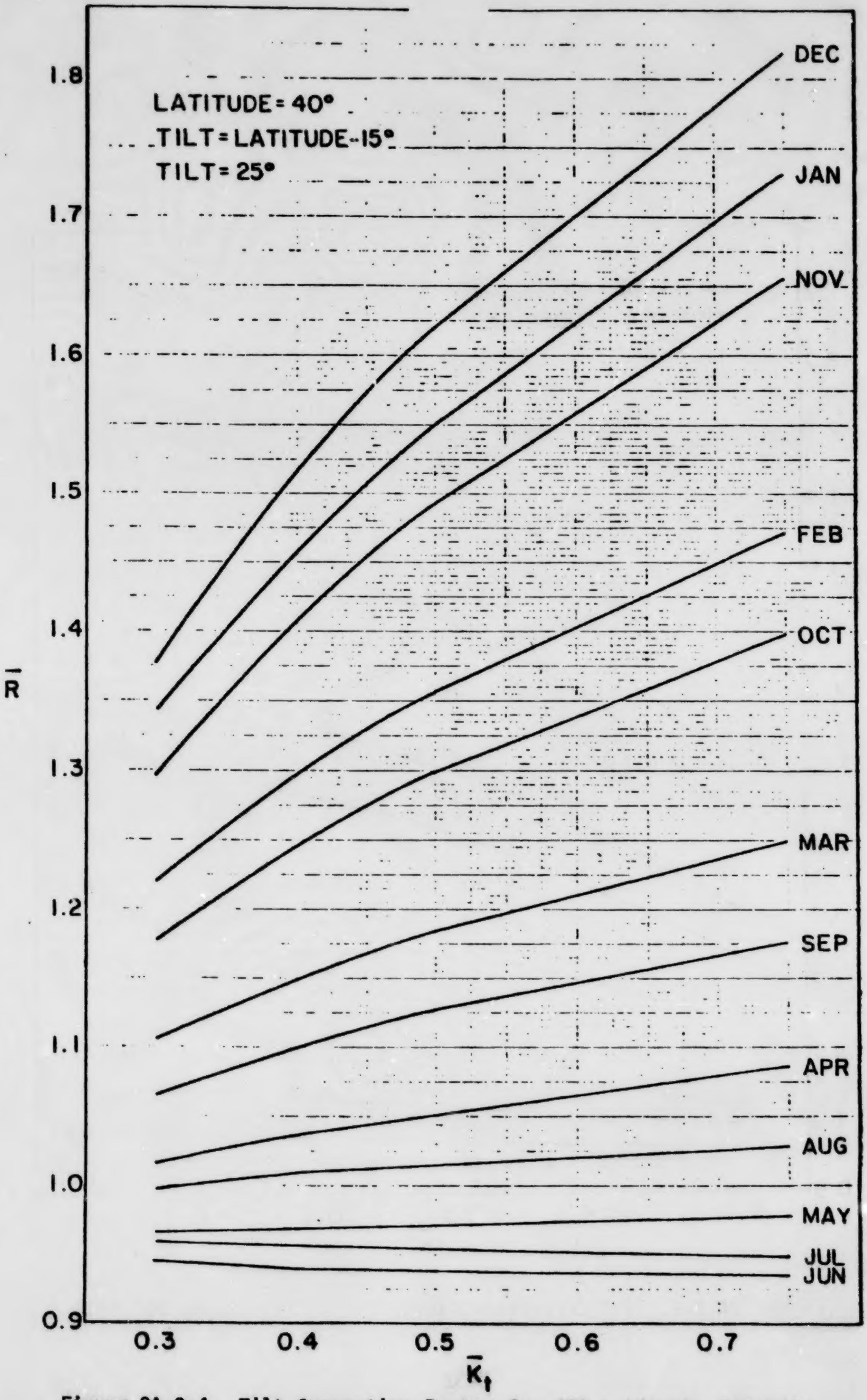

Figure 2A-2-4. Tilt Correction Factor for 40° Latitude, 25° Tilt

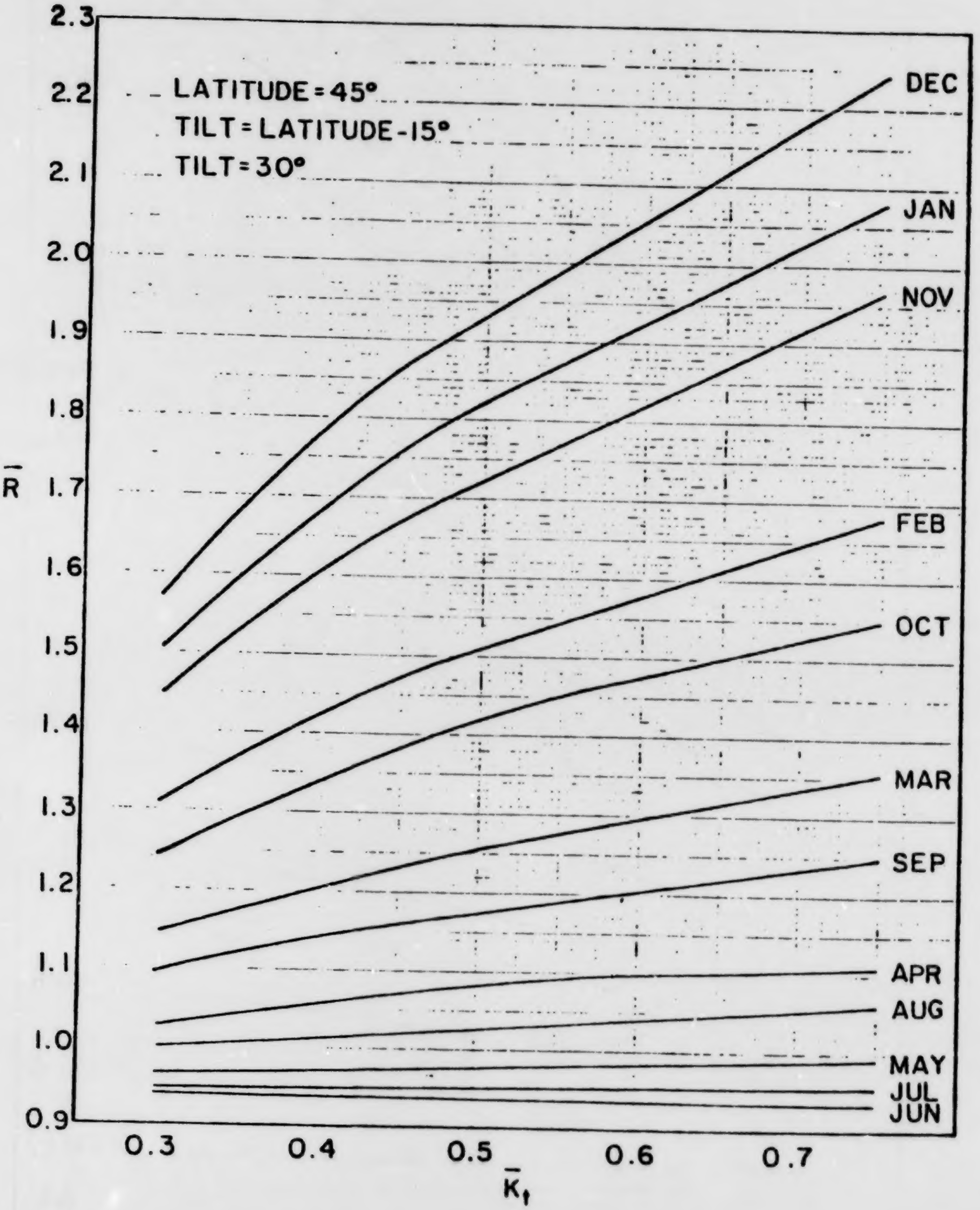

Figure 2A-2-5. Tilt Correction Factor for 45° Latitude, 30° Tilt

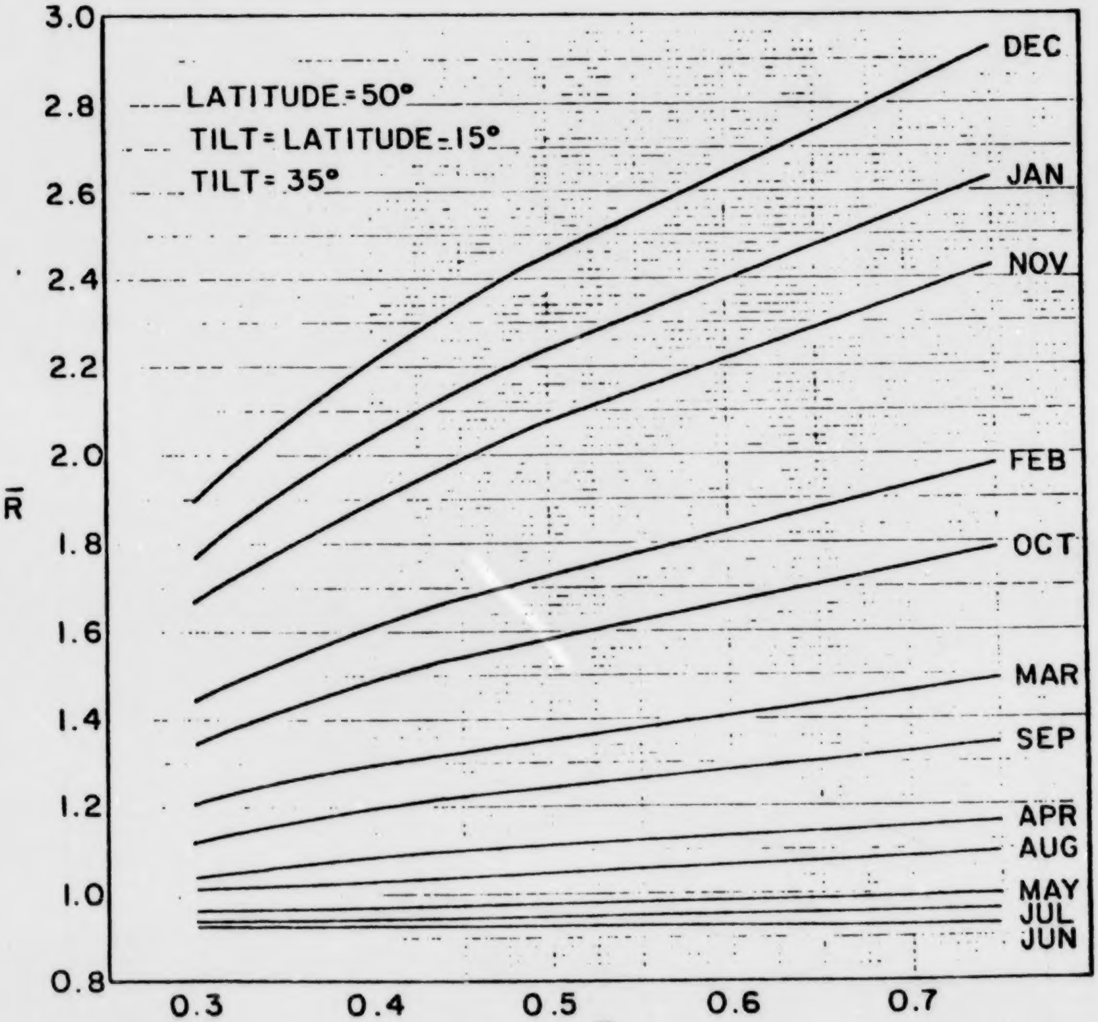

Figure 2A-2-6. Tilt Correction Factor for 50° Latitude, 35° Tilt

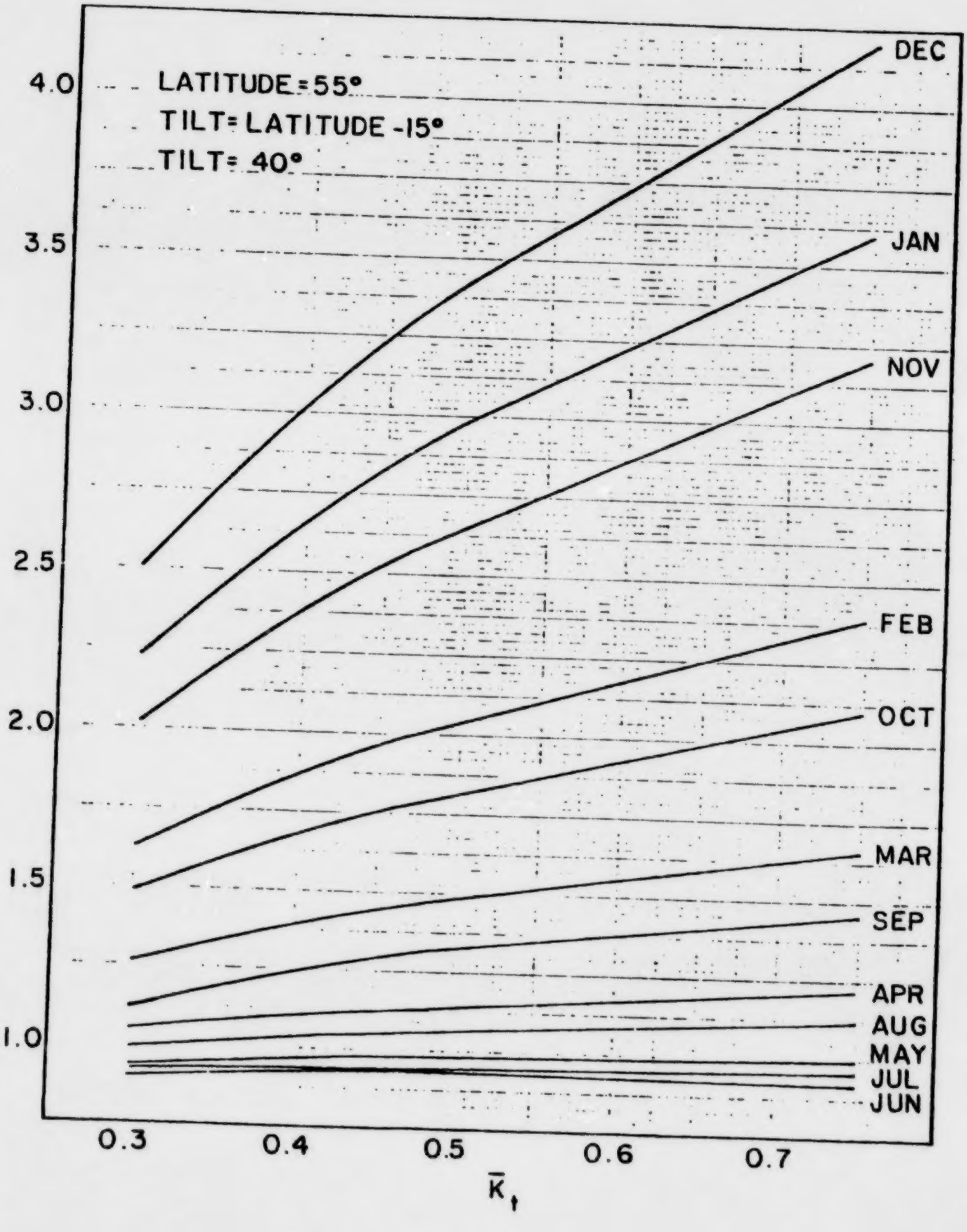

Figure 2A-2-7. Tilt Correction Factor for 55° Latitude, 40° Tilt

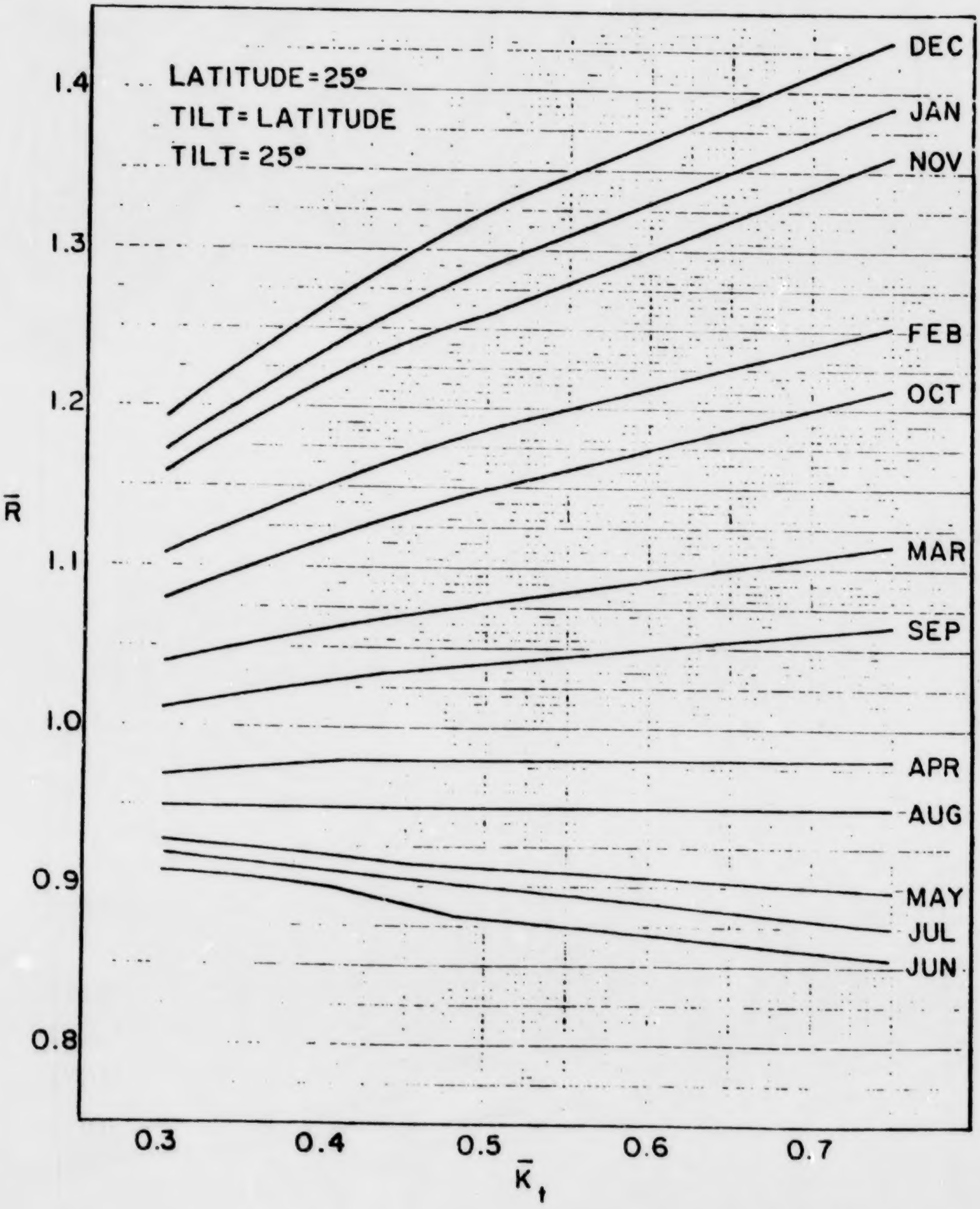

Figure 2A-2-8. Tilt Correction Factor for 25° Latitude, 25° Tilt

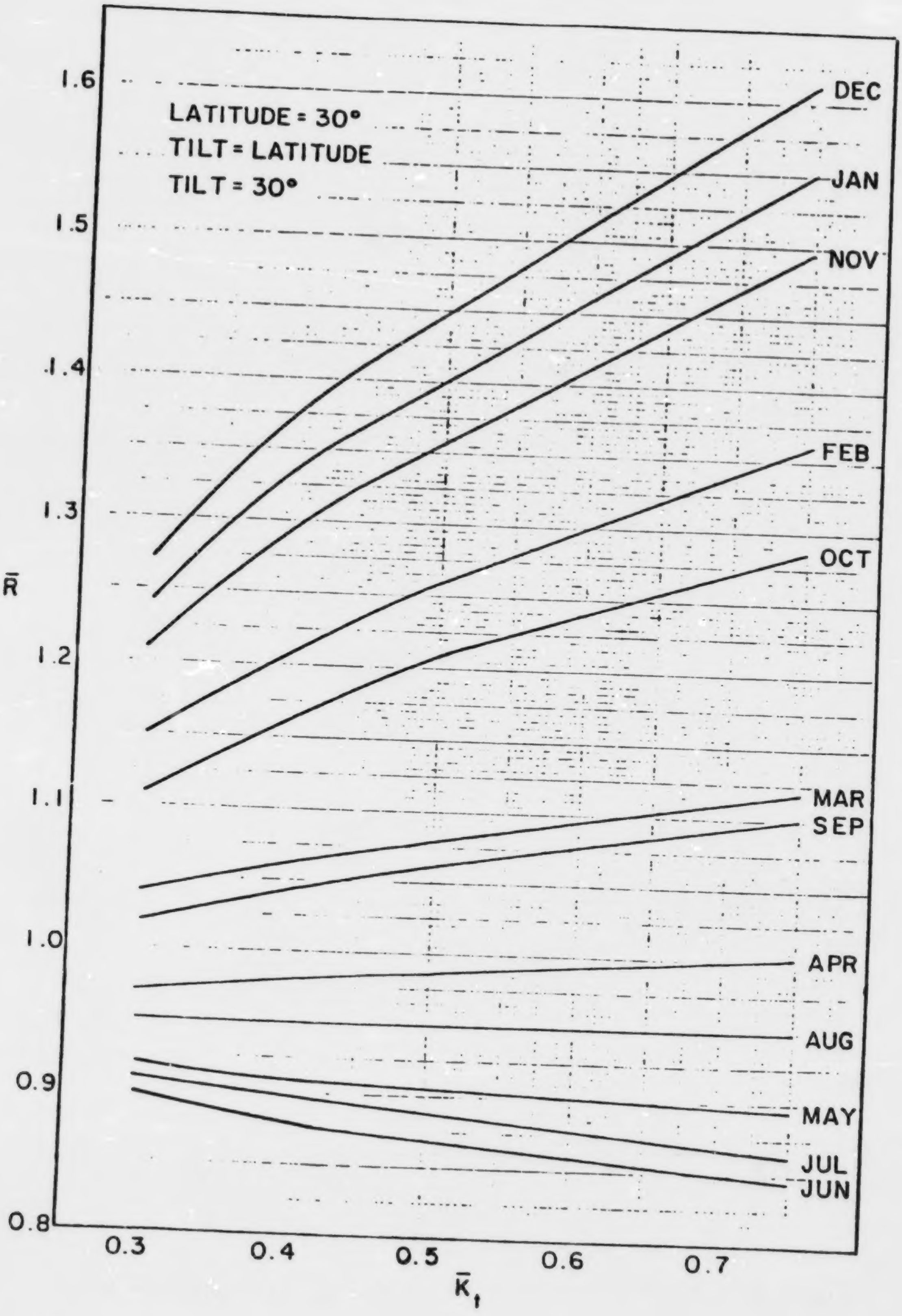

Figure 2A-2-9. Tilt Correction Factor for 30° Latitude, 30° Tilt

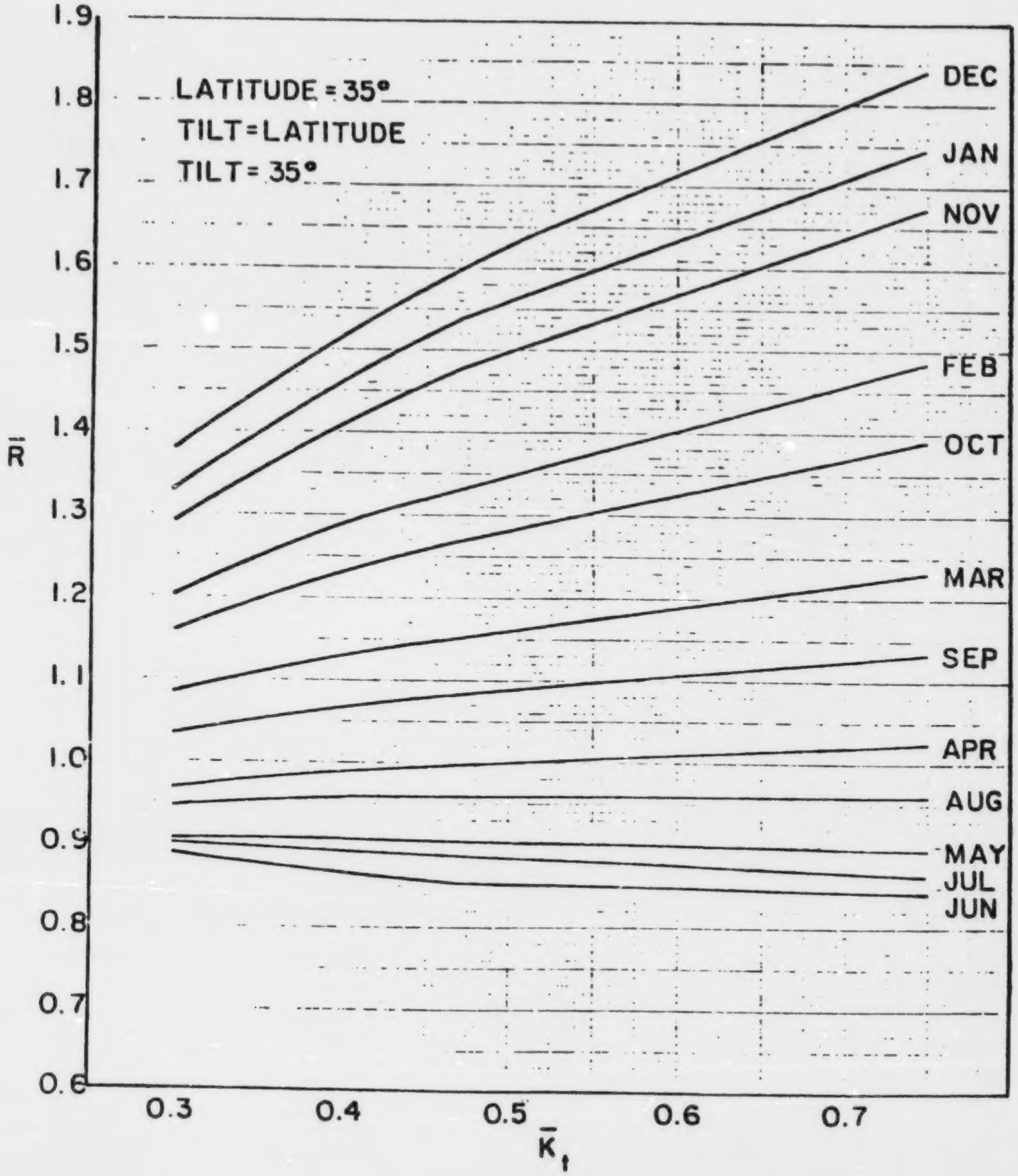

Figure 2A-2-10. Tilt Correction Factor for 35° Latitude, 35° Tilt

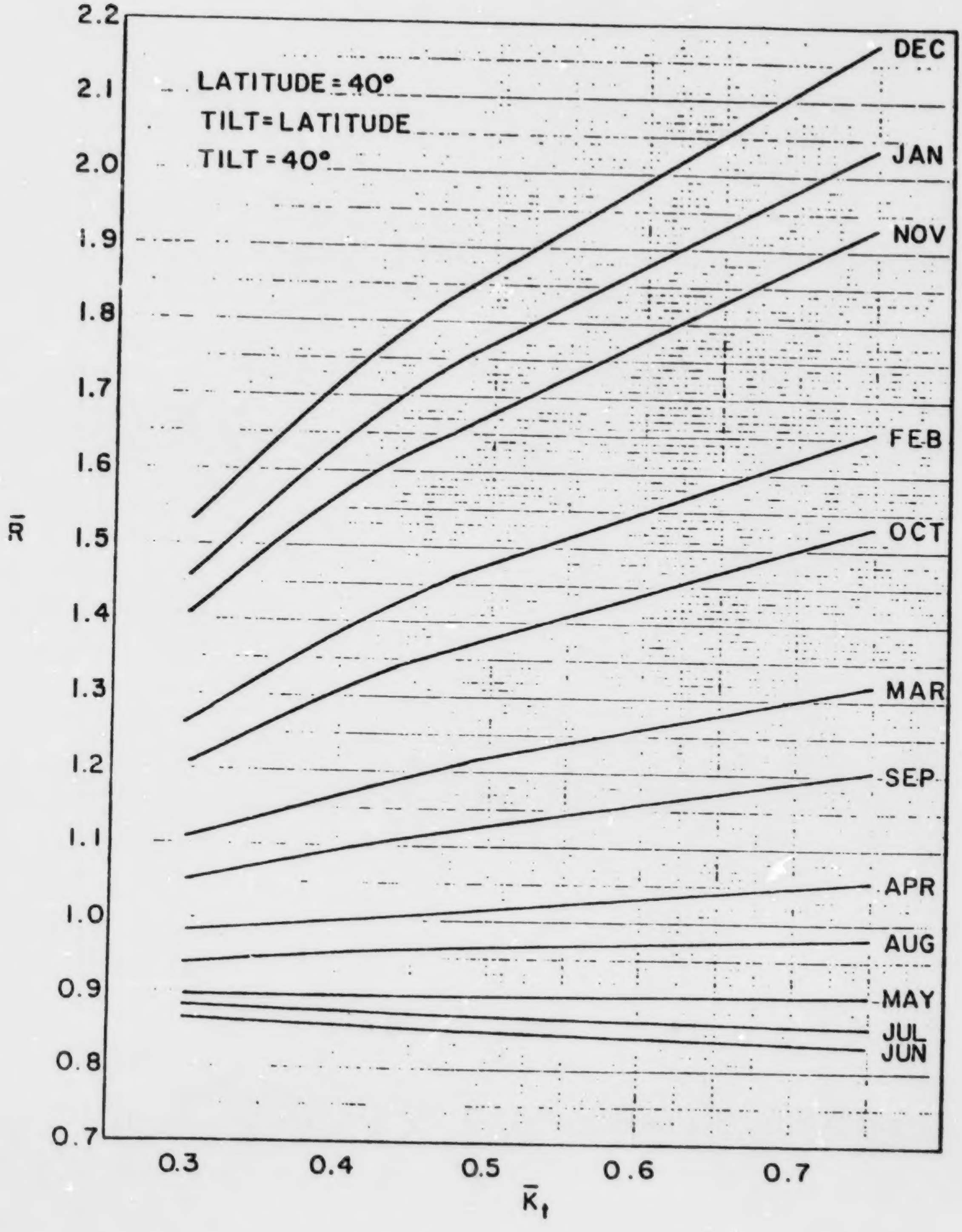

Figure 2A-2-11. Tilt Correction Factor for 40° Latitude, 40° Tilt

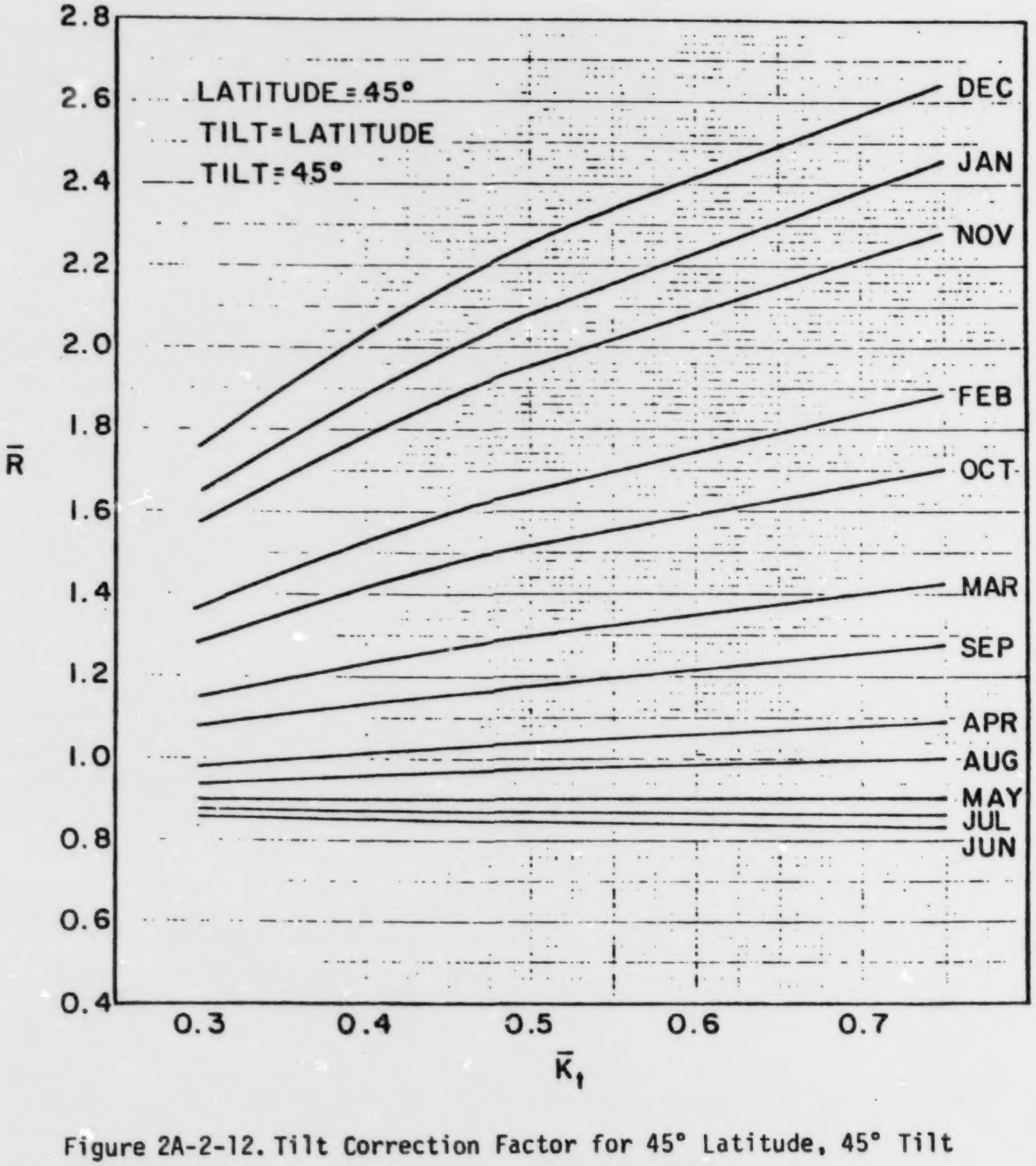

Figure 2A-2-12. Tilt Correction Factor for 45° Latitude, 45° Tilt

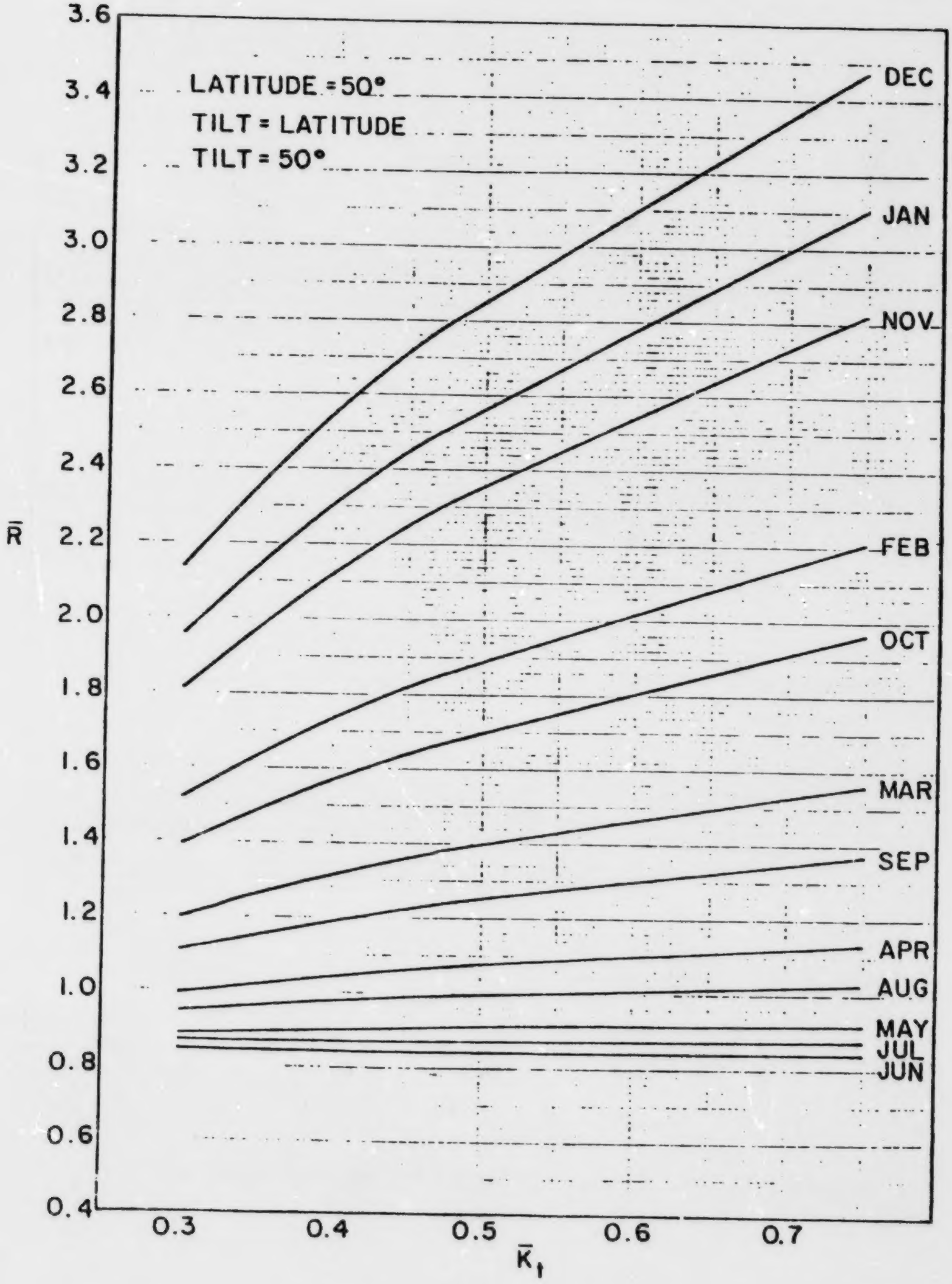

Figure 2A-2-13. Tilt Correction Factor for 50° Latitude, 50° Tilt

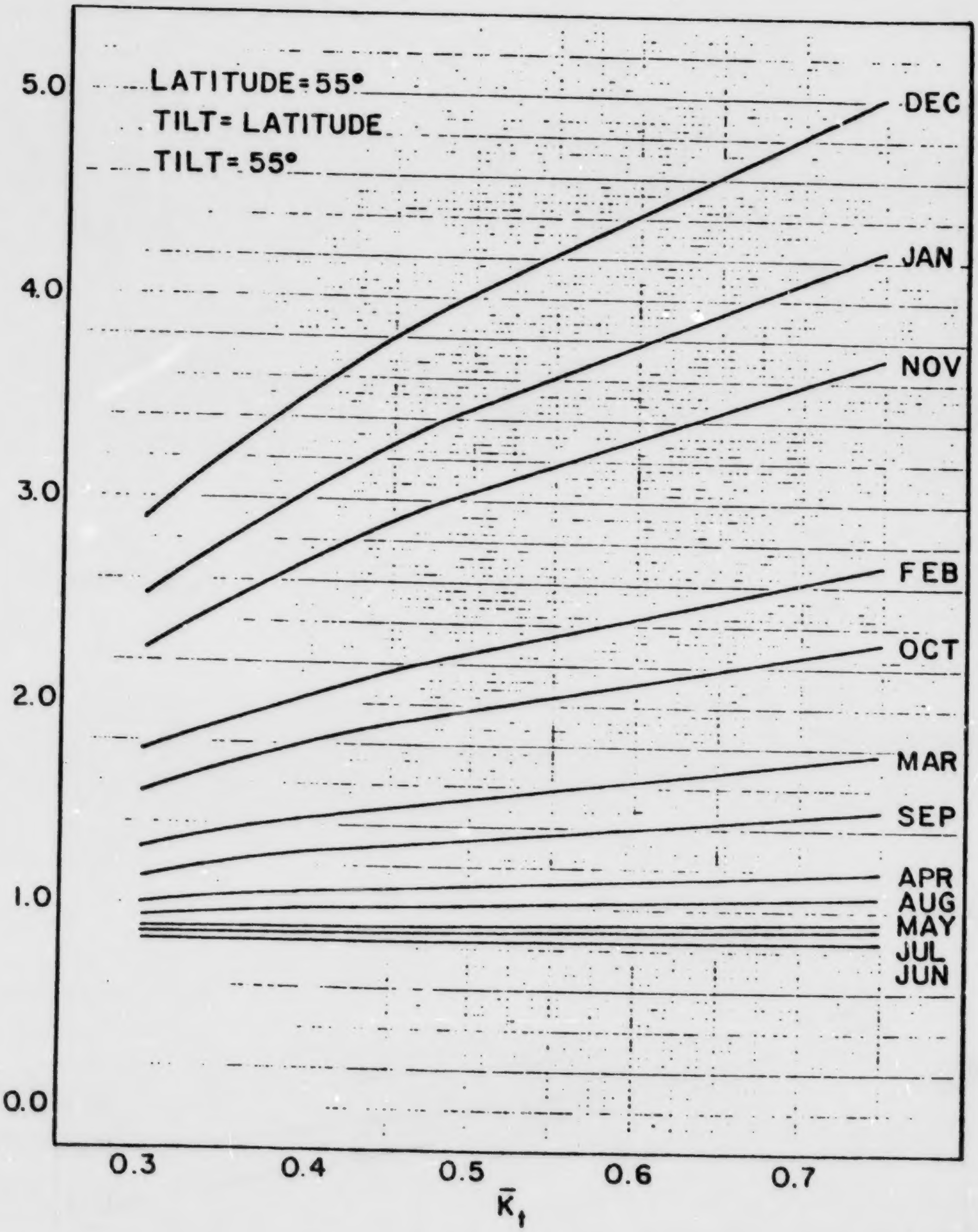

Figure 2A-2-14. Tilt Correction Factor for 55° Latitude, 55° Tilt

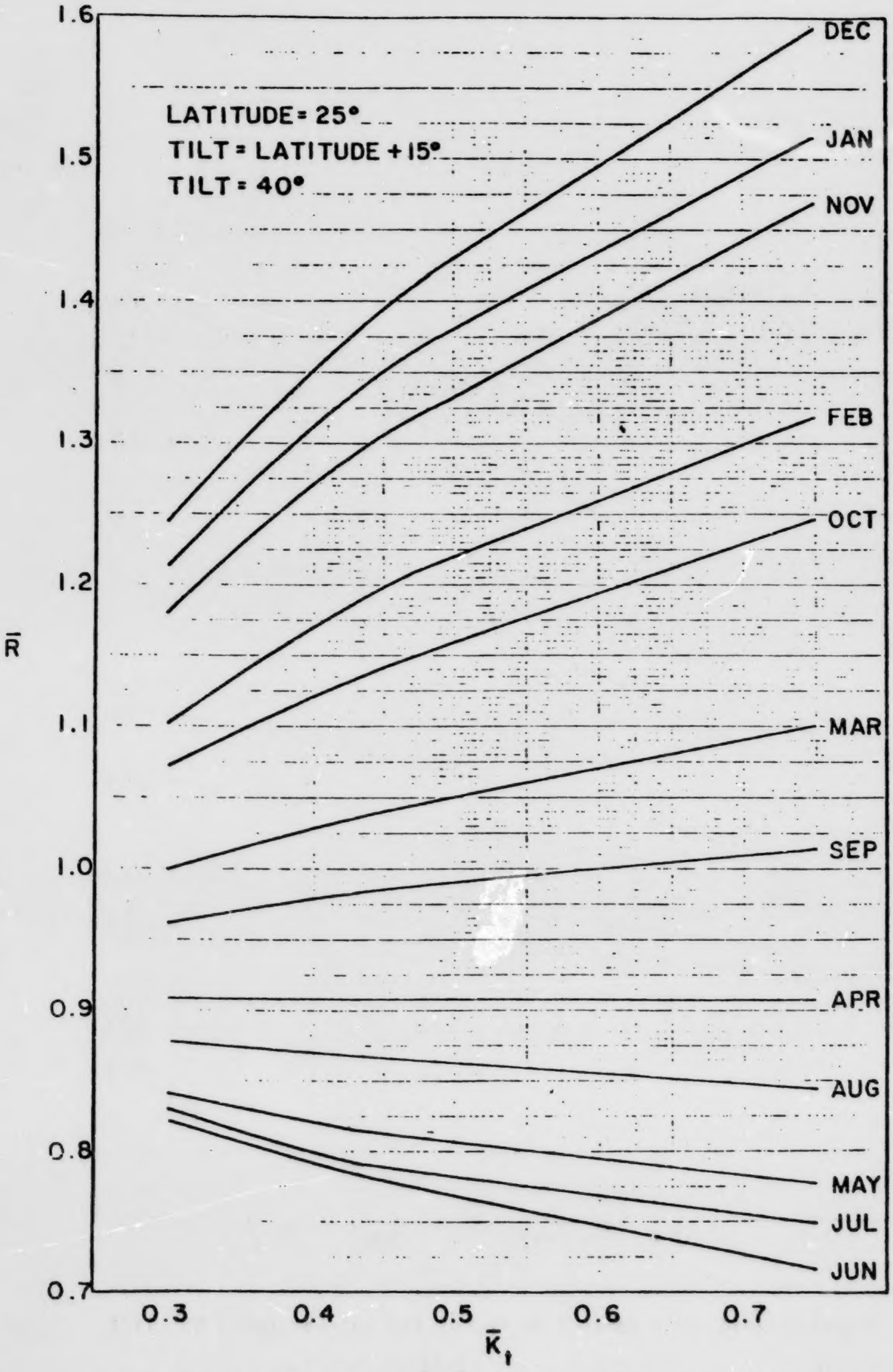

Figure 2A-2-15. Tilt Correction Factor for 25° Latitude, 40° Tilt

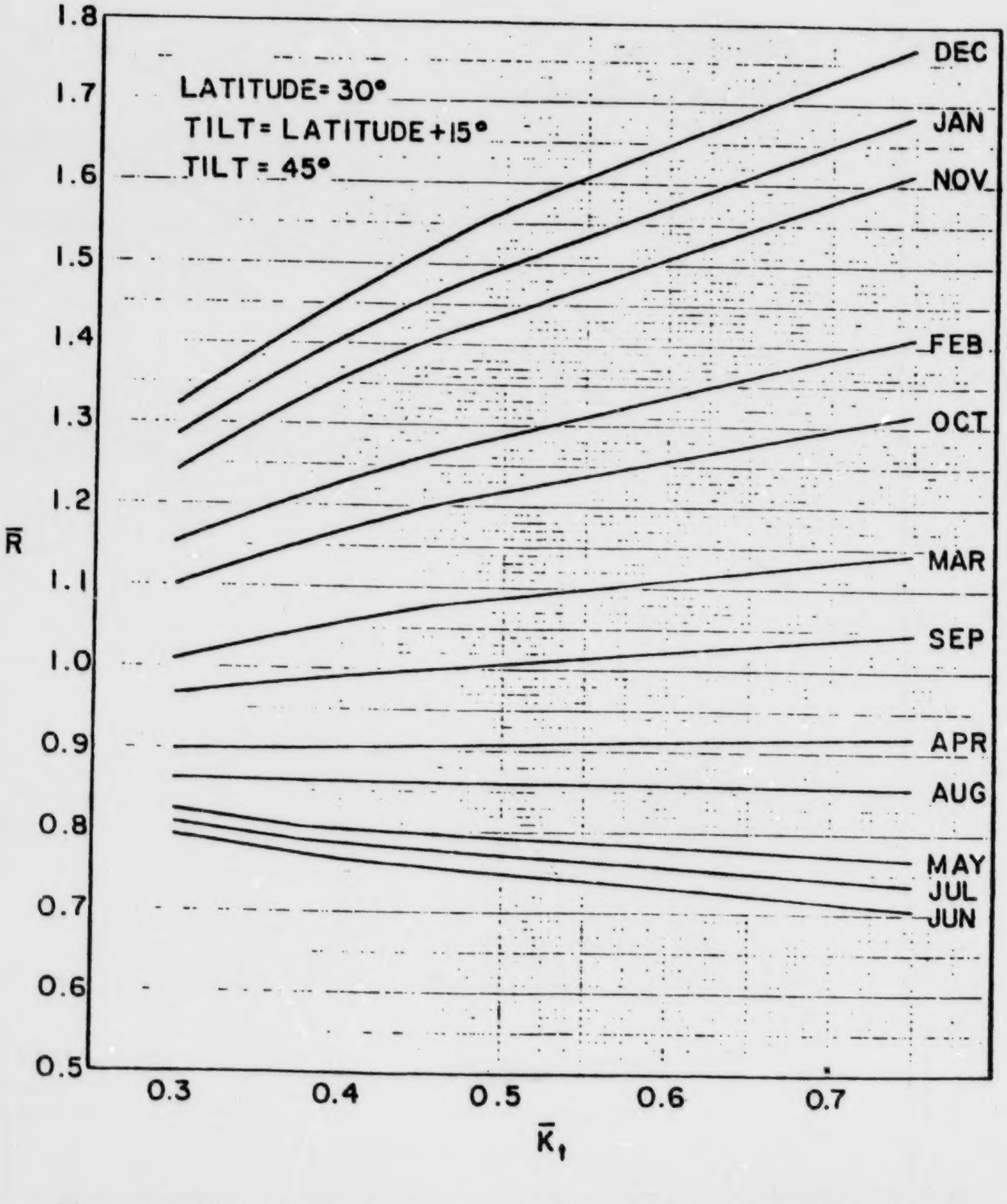

Figure 2A-2-16. Tilt Correction Factor for 30° Latitude, 45° Tilt

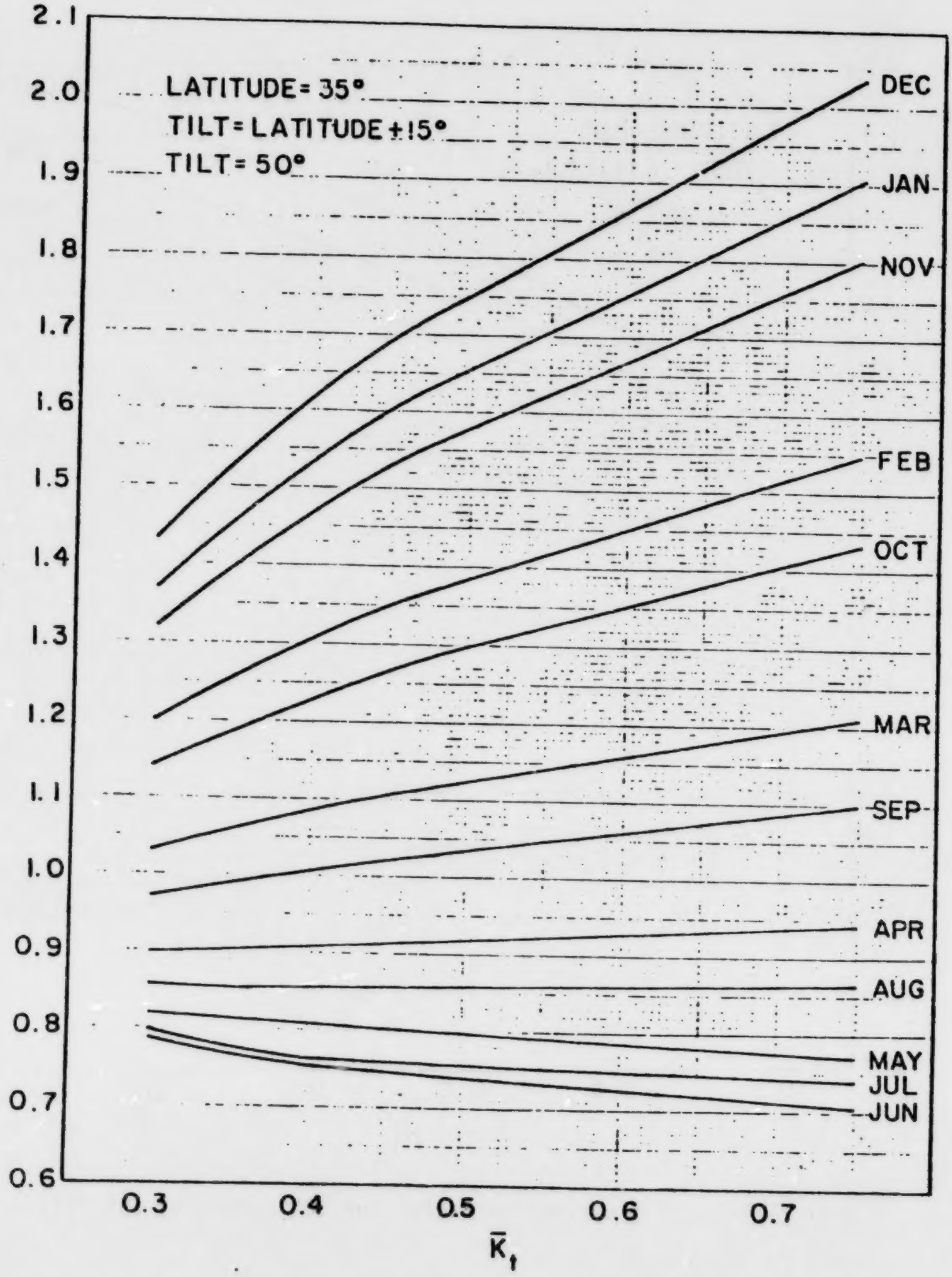

Figure 2A-2-17. Tilt Correction Factor for 35° Latitude, 50° Tilt

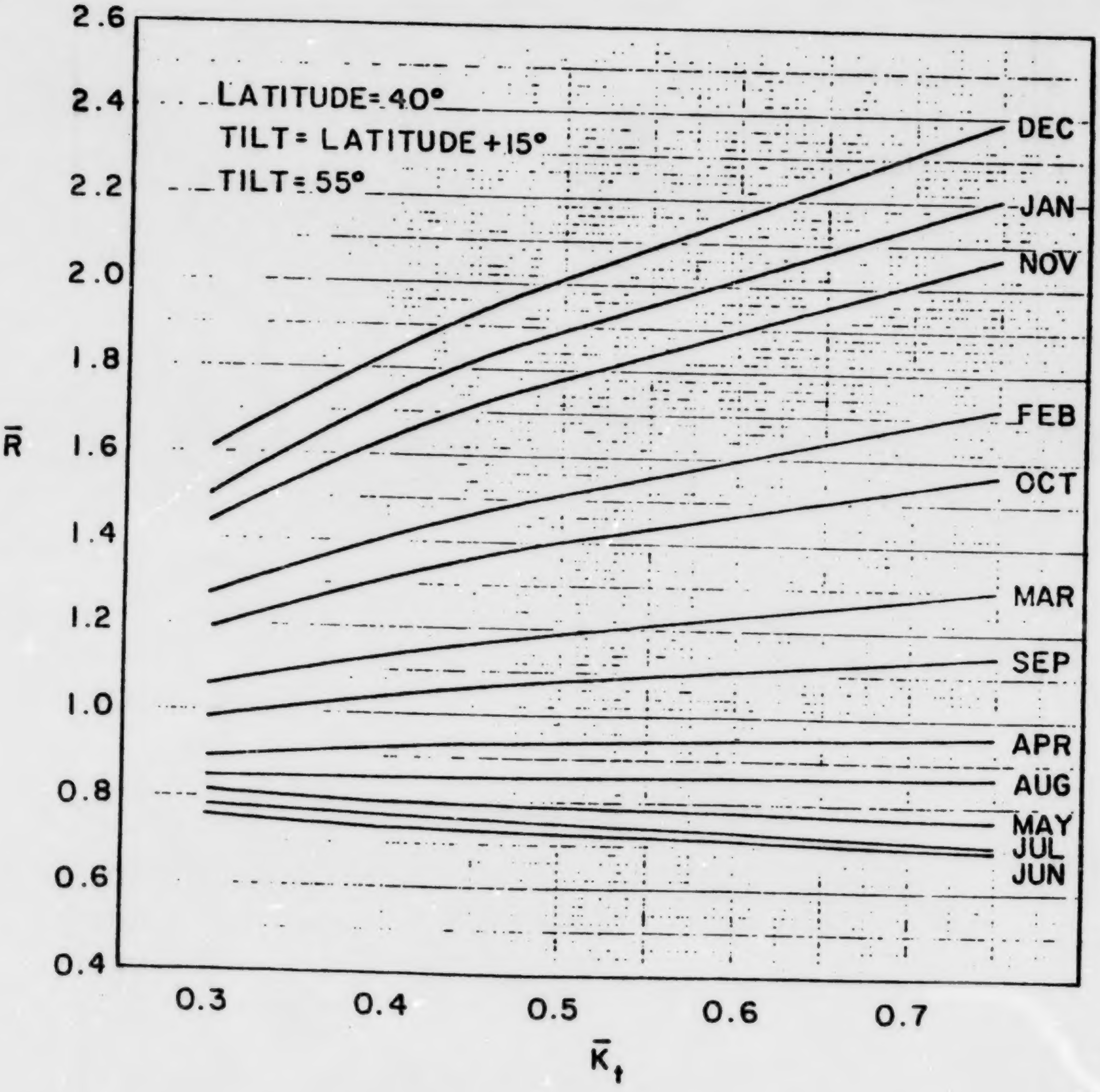

Figure 2A-2-18. Tilt Correction Factor for 40° Latitude, 55° Tilt

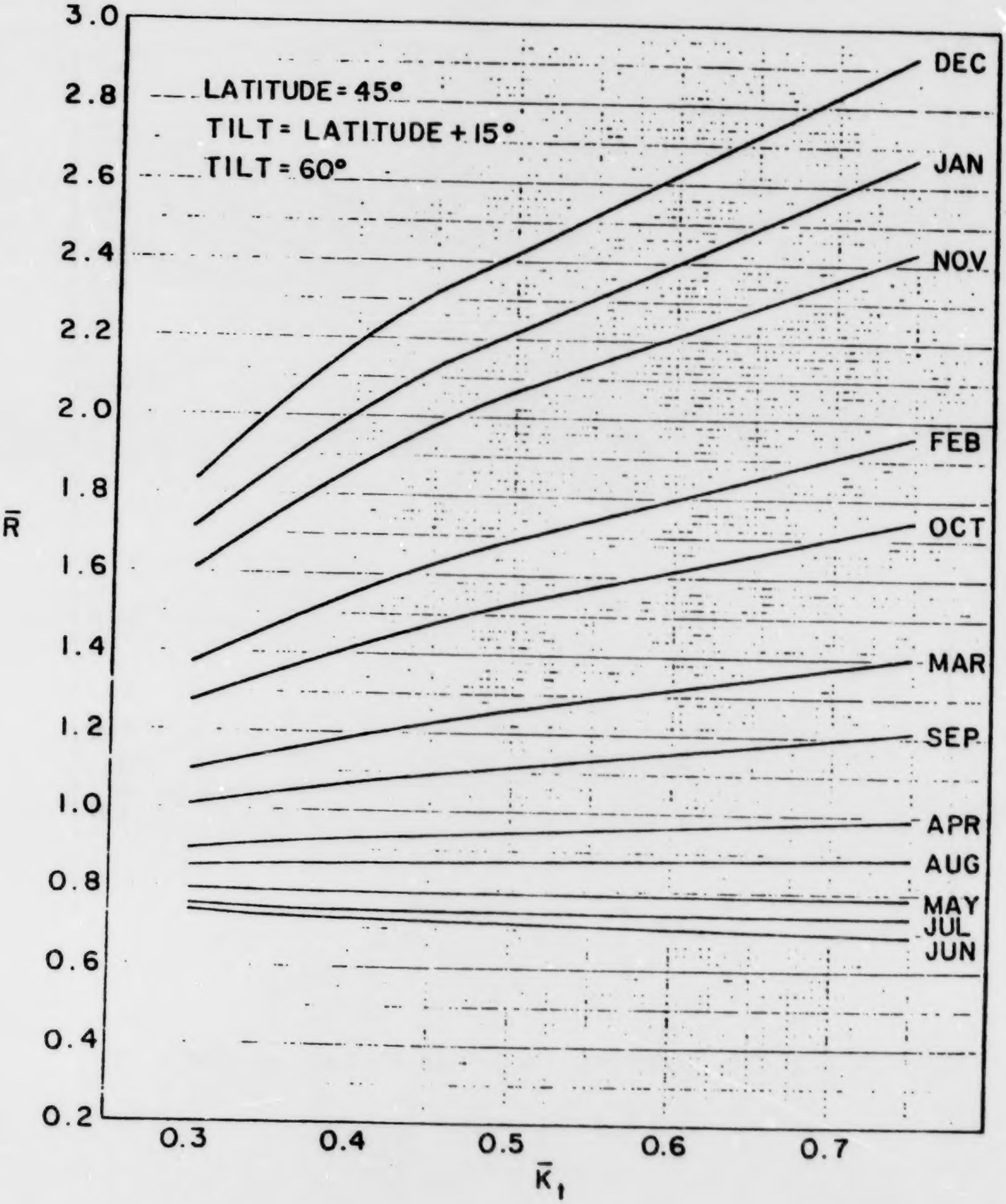

Figure 2A-2-19. Tilt Correction Factor for 45° Latitude, 60° Tilt

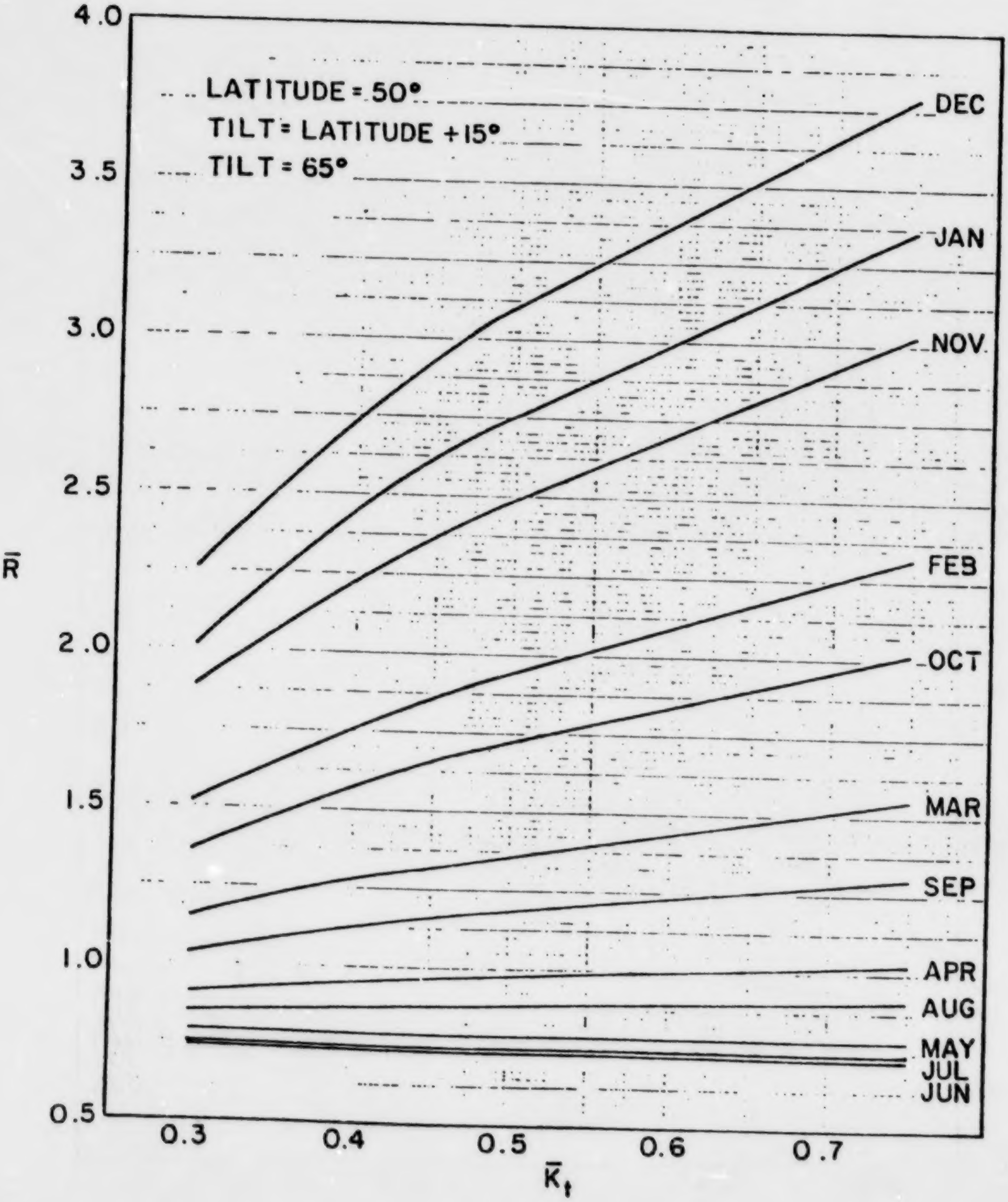

Figure 2A-2-20. Tilt Correction Factor for 50° Latitude, 65° Tilt

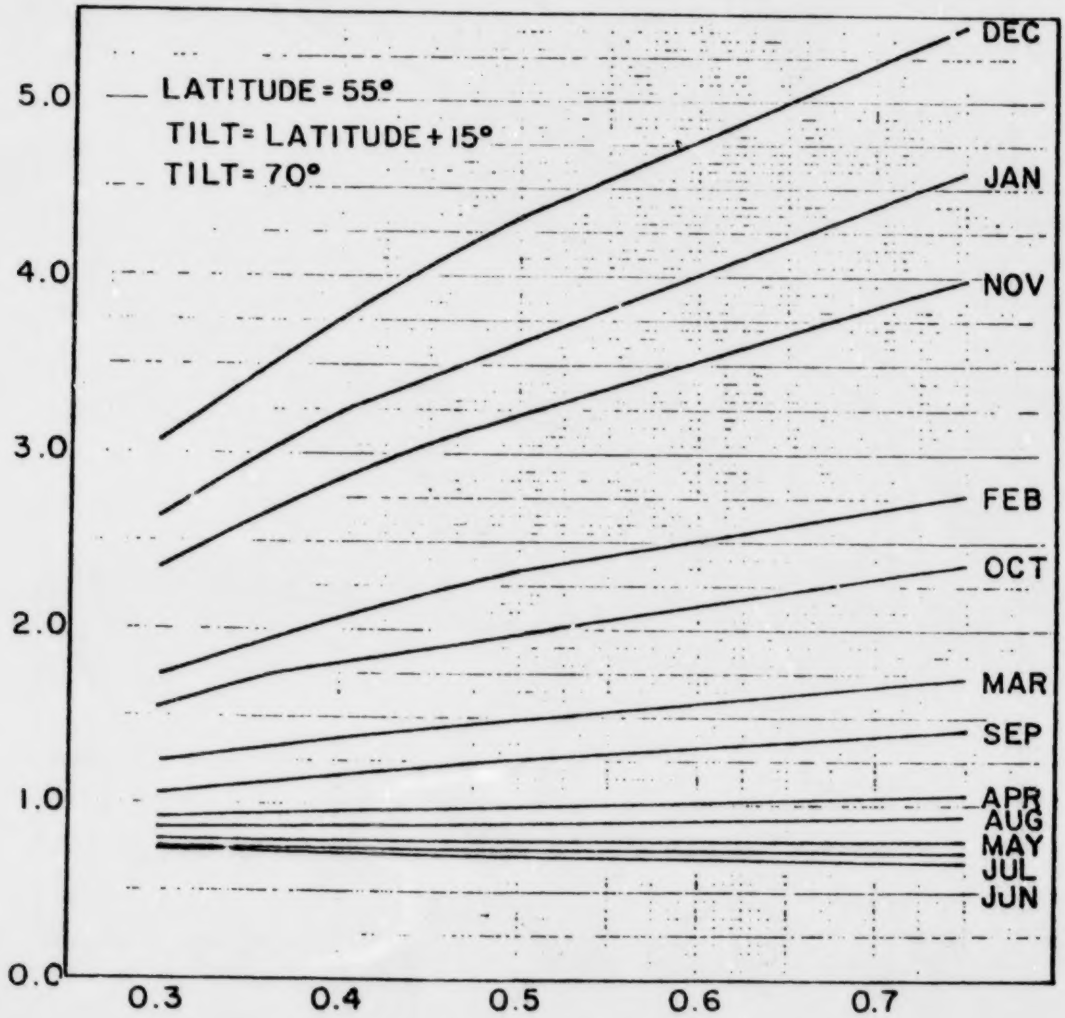

Figure 2A-2-21. Tilt Correction Factor for 55° Latitude, 70° Tilt

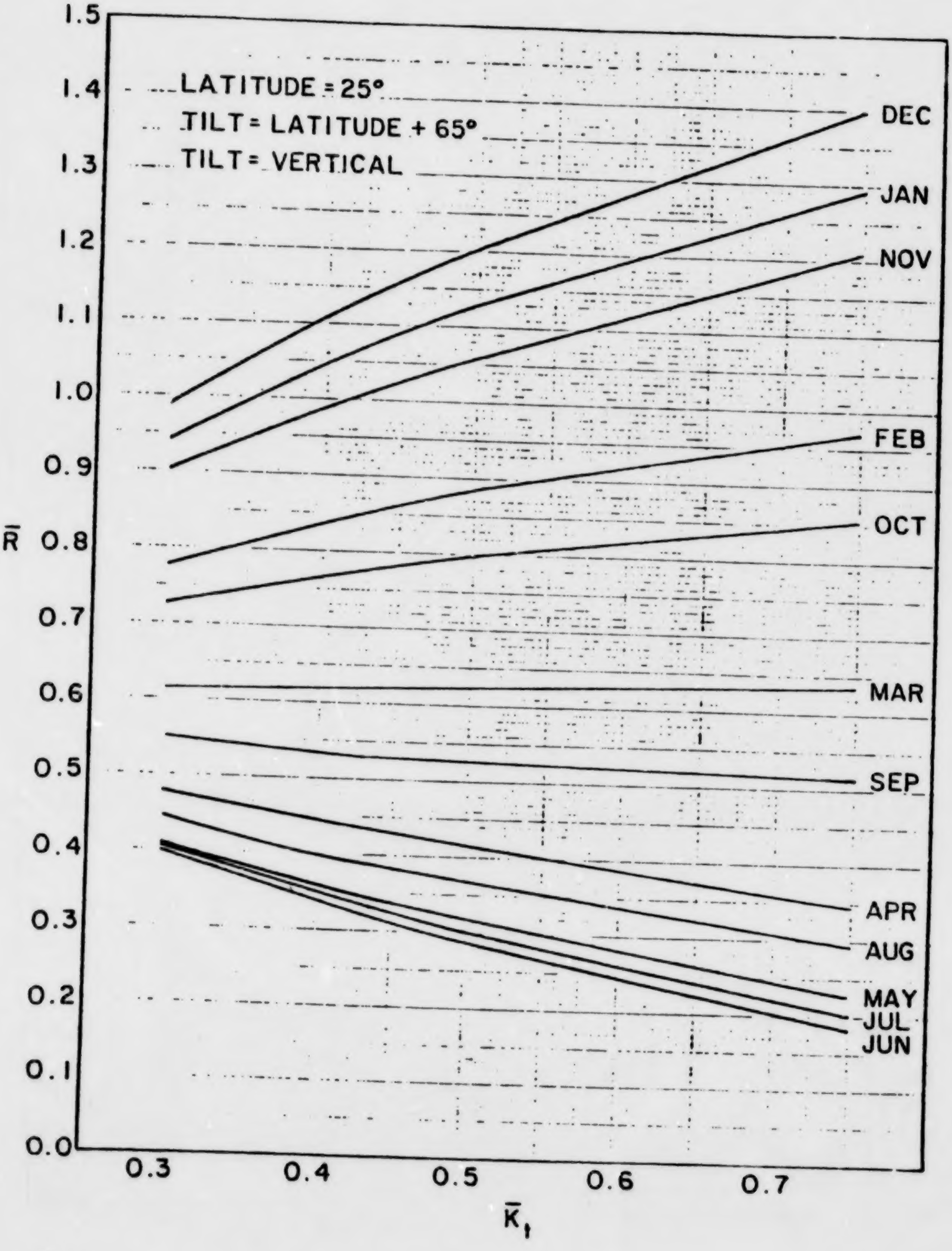

Figure 2A-2-22. Tilt Correction Factor for 25° Latitude, Vertical Tilt

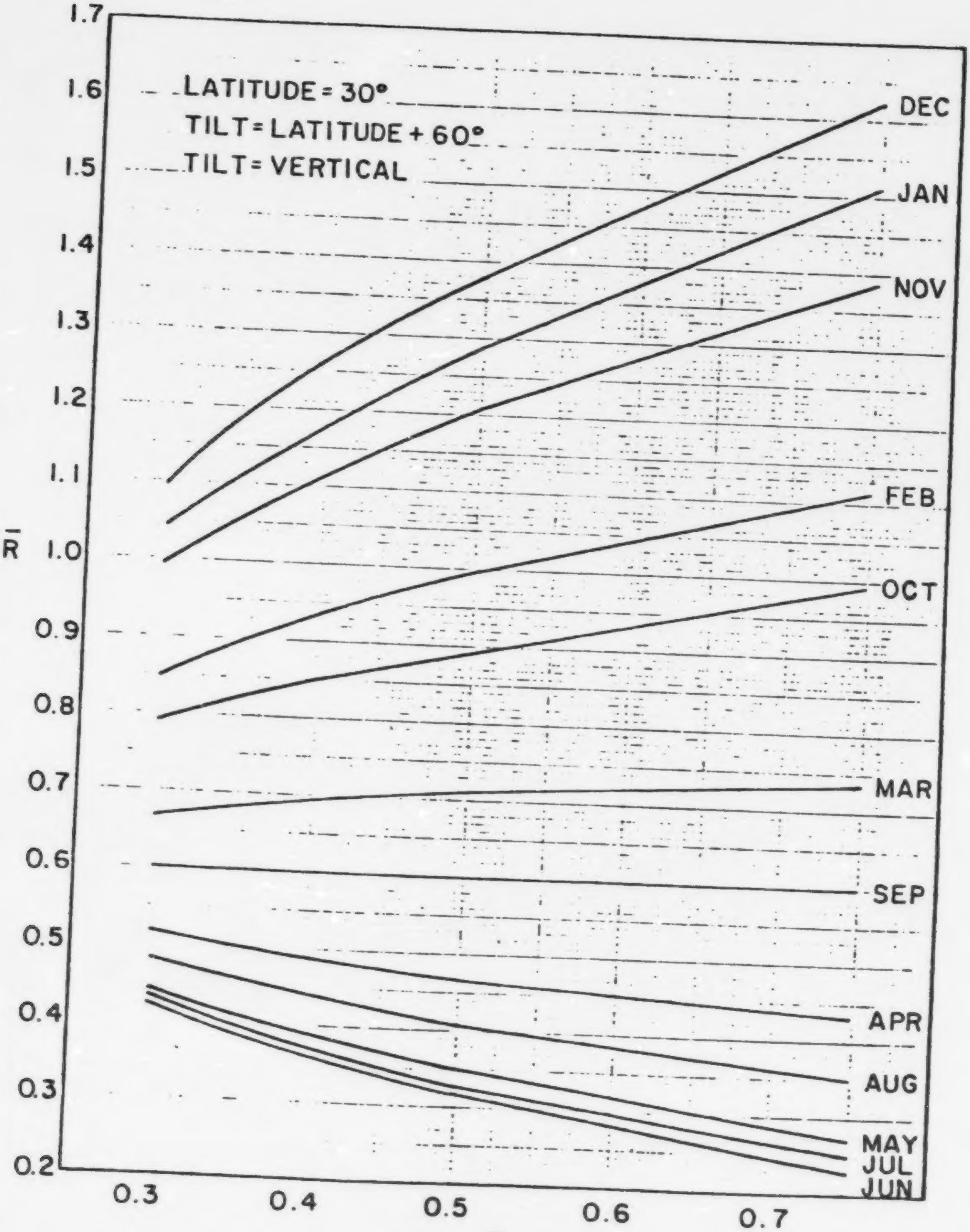

Figure 2A-2-23. Tilt Correction Factor for 30° Latitude, Vertical Tilt

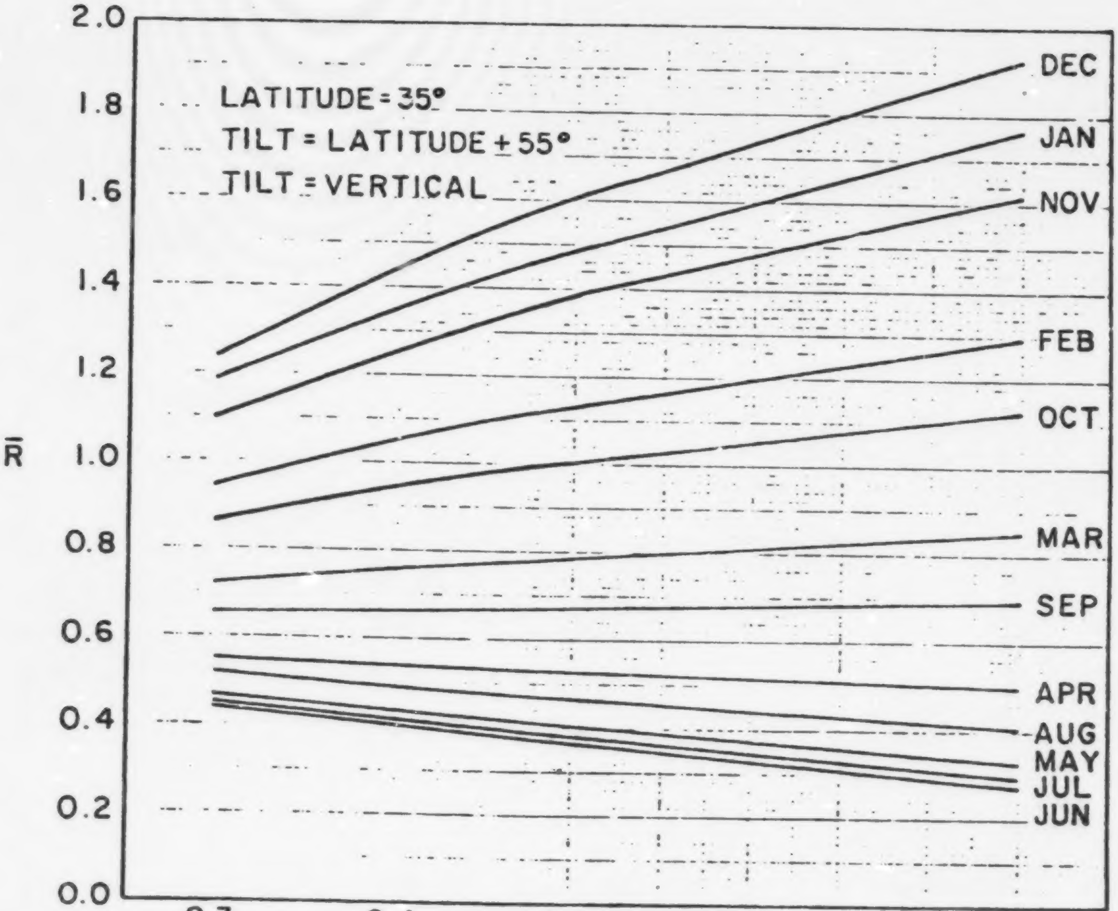

Figure 2A-2-24. Tilt Correction Factor for 35° Latitude, Vertical Tilt

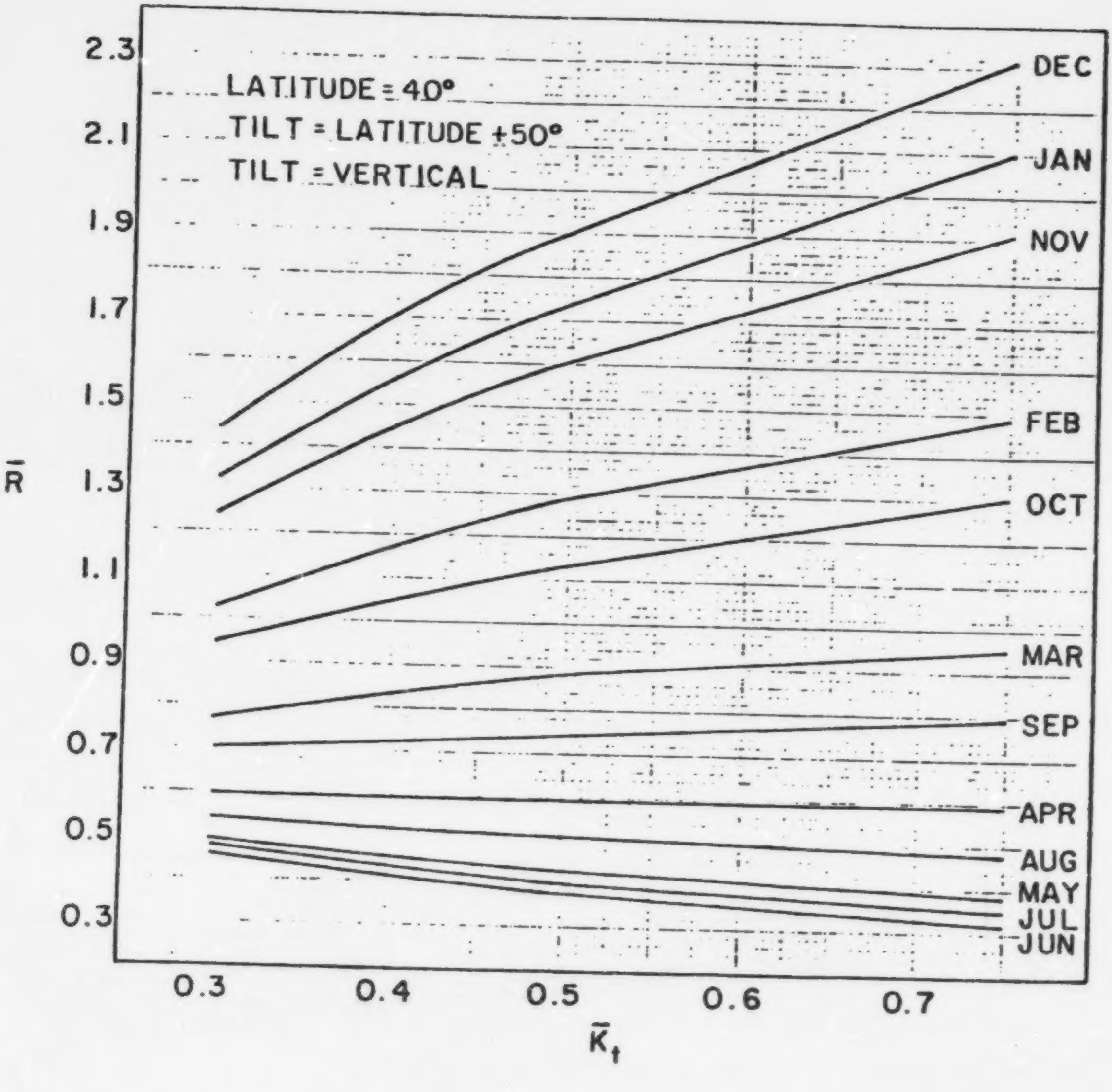

Figure 2A-2-25. Tilt Correction Factor for 40° Latitude, Vertical Tilt

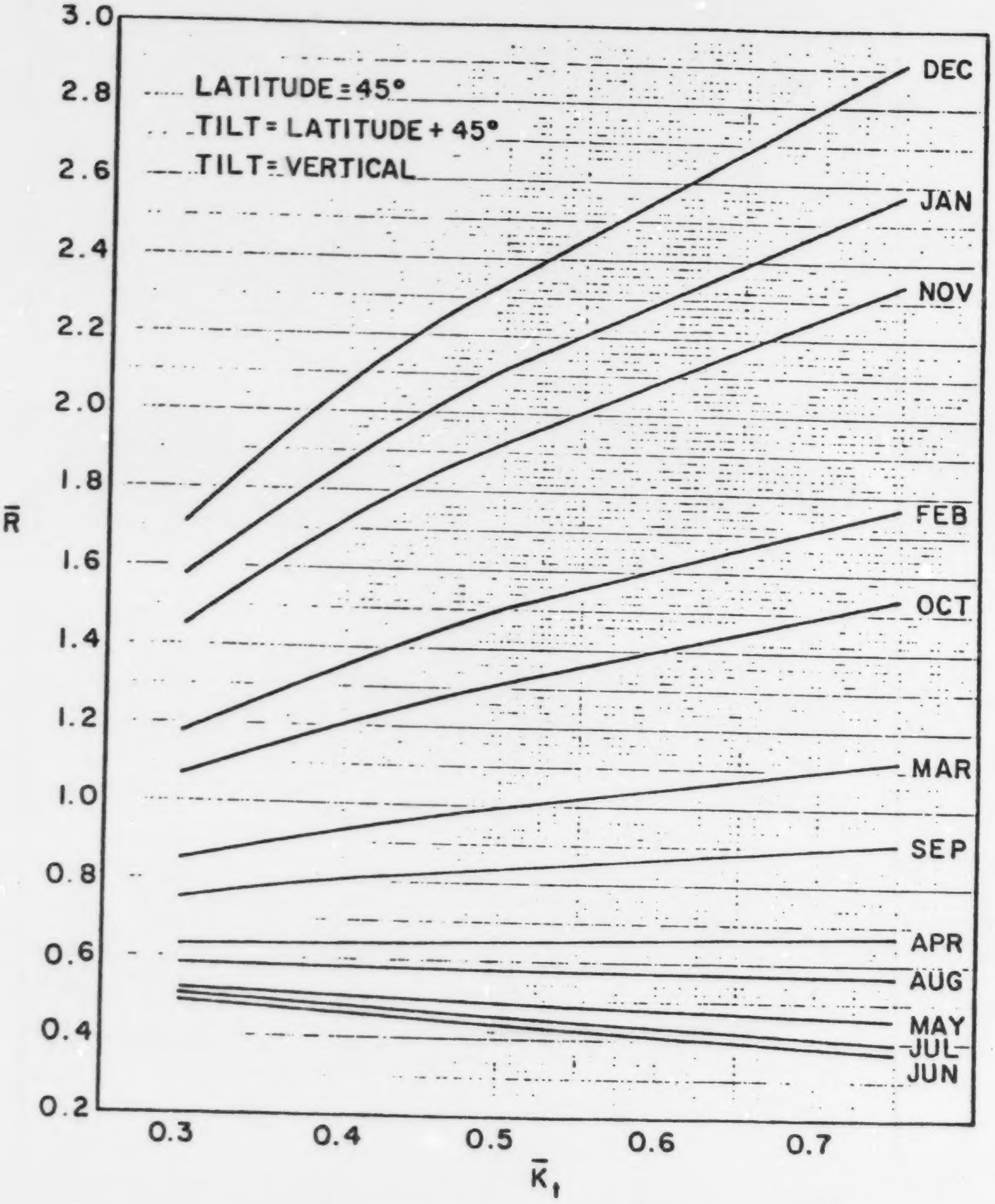

Figure 2A-2-26. Tilt Correction Factor for 45° Latitude, Vertical Tilt

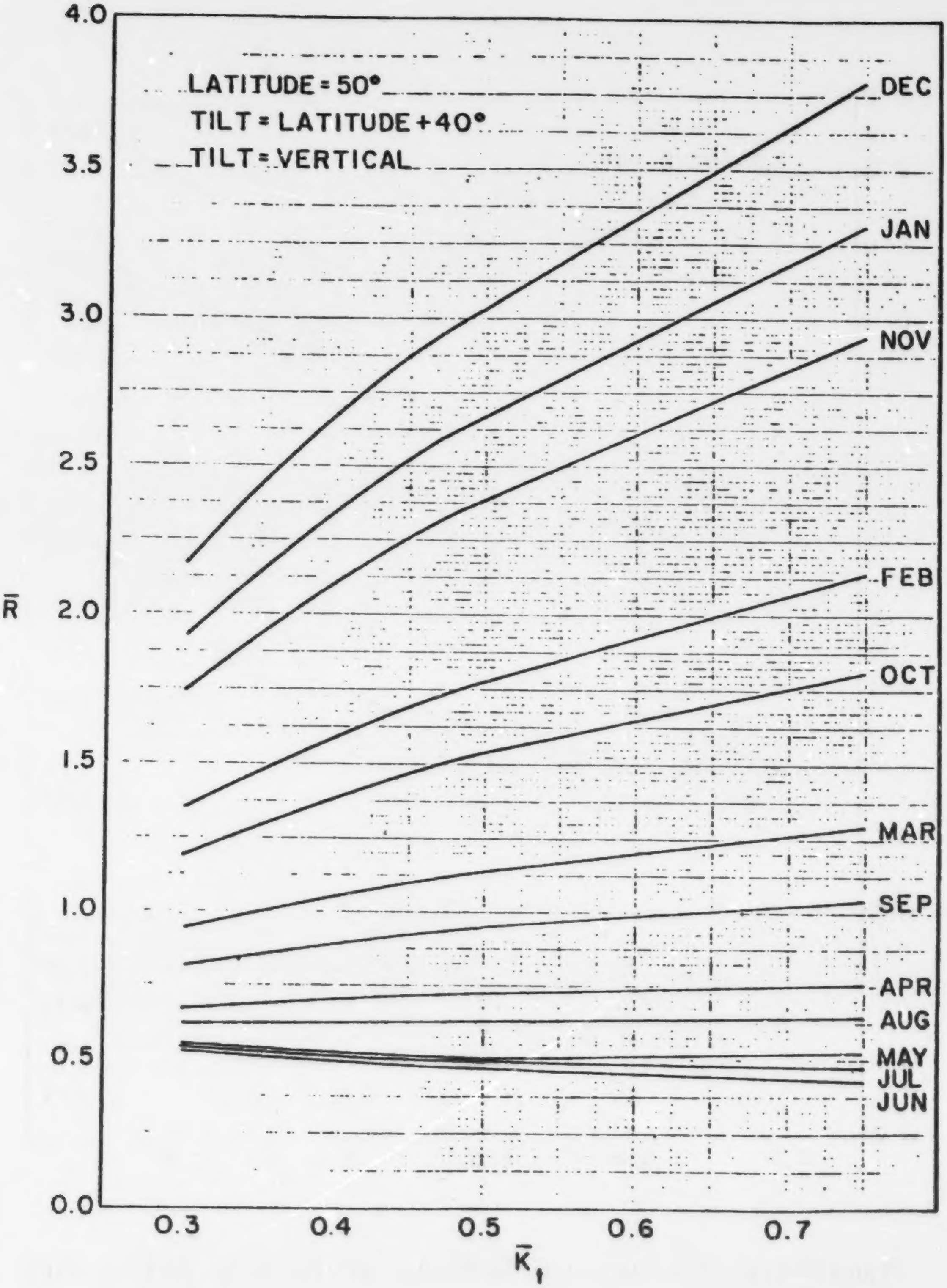

Figure 2A-2-27. Tilt Correction Factor for 50° Latitude, Vertical Tilt

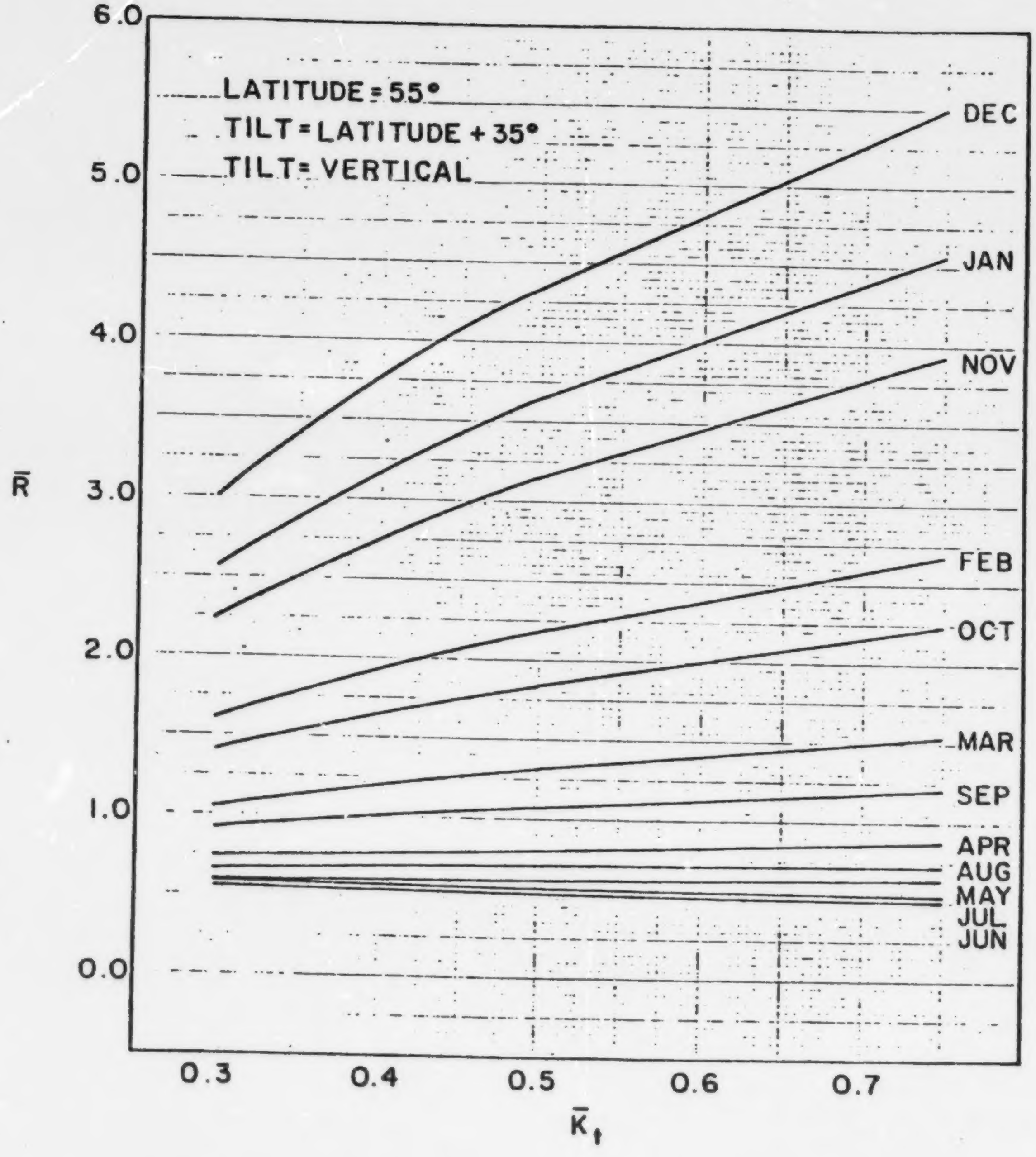

Figure 2A-2-28. Tilt Correction Factor for 55° Latitude, Vertical Tilt

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CHAPTER 3

HEAT TRANSFER AND THE EQUATIONS OF SOLAR COLLECTOR PERFORMANCE

3.1 INTRODUCTION

The topics covered in this chapter are:

1. The transmission of radiation through collector covers, and absorbtion by absorber surfaces.
2. Heat transfer in flat plate collectors.
3. The collector efficiency equation.

The material presented in this chapter is covered in more detail in Chapters 5, 6 and 7 of D and B. The objective of these notes is to provide the student with a general understanding of the thermal processes occurring in a flat plate collector, not to develop his capability to carry out quantitative analysis of these processes. The analysis presented is for a liquid cooled tube in plate collector of conventional design, but the same approach can be applied to other collector types.

3.2 TRANSMISSION AND ABSORBTION IN FLAT PLATE COLLECTORS

3.2.1 Transmission Through Partially Transparent Media

Figure 3.2.1 shows a beam of radiation passing from a medium with refractive index n_1 to one with refractive index n_2

Snell's law

$$\frac{n_1}{n_2} = \frac{\text{Sin } \Theta_2}{\text{Sin } \Theta_1} \qquad \text{(Eqn 3.2.1)}$$

relates the angles to the refractive indices.

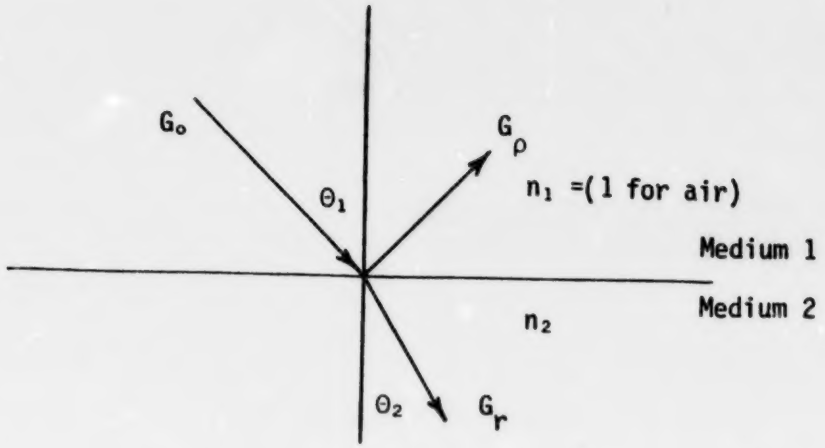


FIGURE 3.2.1

REFLECTION AT A SURFACE

The reflectance of the surface is a function of n_1, n_2, & Θ_1. Fresnel's law, which describes this relationship, is

$$r(o) = \frac{G_r}{G_i} = \left(\frac{n_1 - n_2}{n_1 - n_2}\right)^2 \qquad \text{(Eqn 3.2.2)}$$

for $\Theta_1 = 0$, the relationship simplifies to

$$r(o) = \left(\frac{n - 1}{n + 1}\right)^2 \qquad \text{(Eqn 3.2.3)}$$

where the upper medium has n = 1 (a good approximation for air)

For glass, n = 1.526 (averaged over the solar spectrum) and

$$r(o) = \left(\frac{.526}{2.526}\right)^2 = .0434$$

For an incident angle of 60° from the normal, Fresnel's equation would yield

$$r(60) = 0.93$$

A cover glass has two surfaces, both reflecting. For normal incidents the reflectances are the same and the reflected and transmitted beam strengths arc as shown in Figure 3.2.2.

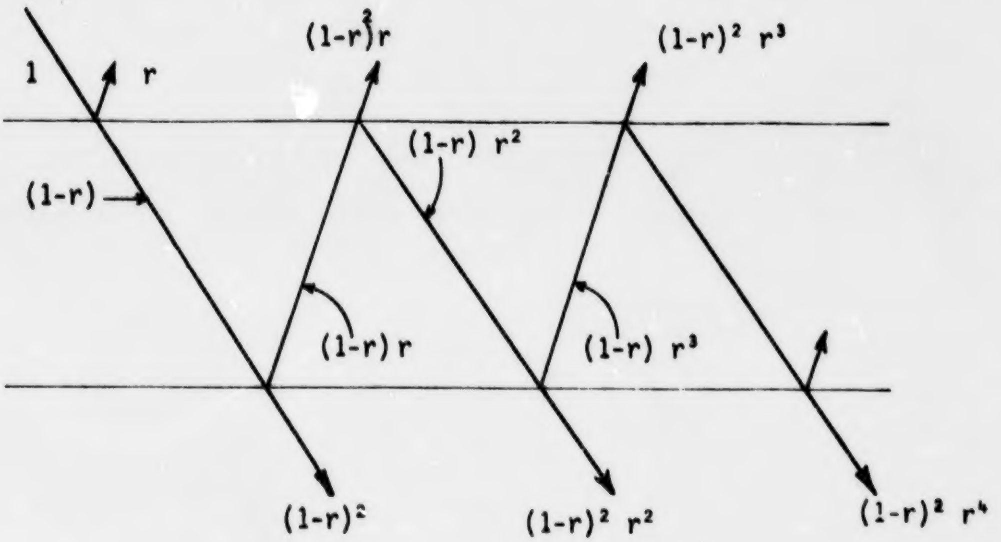

1
r
$(1-r)^2 r$
$(1-r)^2 r^3$
$(1-r)$
$(1-r)\ r^2$
$(1-r)\ r$
$(1-r)\ r^3$
$(1-r)^2$
$(1-r)^2\ r^2$
$(1-r)^2\ r^4$

FIGURE 3.2.2

TRANSMISSION THROUGH

ONE NON-ABSORBING COVER

Summing the transmitted terms yields

$$\tau_r = (1-r)^2 \sum_{j=0}^{\infty} r^{2j} = \frac{1-r}{1+r} \qquad \text{(Eqn 3.2.4)}$$

with N collector covers

$$\tau_{r,N} = \frac{1-r}{1+(2N-1)\ r} \qquad \text{(Eqn 3.2.5)}$$

for normal incidence.

Figure 3.2.3 shows how the transmittance of glass covers varies for other angles of incidence.

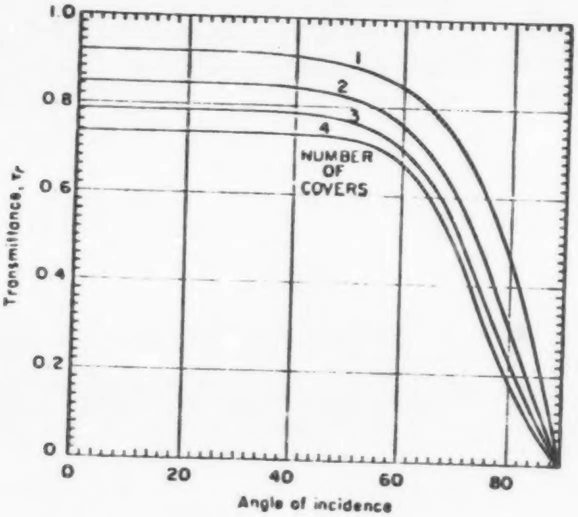


FIGURE 3.2.3

TRANSMITTANCE OF 1,2,3, AND 4 NONABSORBING

COVERS HAVING AN INDEX OF REFRACTION OF 1.526

This analysis assumed no absorbtion in the glass. In fact, the beam strength will decay exponentially and the transmittance considering only absorbtion will be

$$\tau_\alpha = e^{-KL} \qquad \text{(Eqn 3.2.6)}$$

where L = path length and K = extinction coefficient

For glass $.04 < K < .32 \text{ cm}^{-1}$.

As an example, for high quality 0.32 cm (1/8") glass,

$$\tau_\alpha = e^{.04x.32} = .987$$

When $\tau_\alpha \cong 1$, the strengths of the beams in Figure 3.2.2 are only slightly affected by absorbtion, so that the two transmittances can be considered essentially independent and

$$\tau \cong \tau_\alpha \tau_r \qquad \text{(Eqn 3.2.7)}$$

This is a good approximation for collector cover plates. Using the values from our previous examples, a pane of highest quality glass 0.32 cm (1/8") thick will have a transmittance of

$$\begin{aligned} \tau &= \tau_\alpha \tau_r = \tau_\alpha \left(\frac{1-r}{1+r}\right) \\ &= .987 \left(\frac{1-.0434}{1+.0434}\right) \\ &= .905 \end{aligned}$$

for radiation at normal incidence. Table 3.2.1 has other examples.

Figure 3.2.4 shows the transmittances of several plate systems as a function of incident angle.

3.2.2 Absorbtion At The Plate Absorber

If an incident beam of intensity G falls on a solar collector, $I\tau$ will penetrate the cover system and $G\tau\alpha$ will be absorbed. However, the radiation reflected by the absorber is not lost, and may be recovered in subsequent reflections, as shown in Figure 3.2.5.

Table 3.2.1

GLAZING MATERIAL COMPARISON

Property	Teflon	Tedlar	Mylar	Lexan	Kalwal	PPG Herculite	ASG Tempered
Solar transmission (%)	96	92-94	85	82-89	85.90	85	91
Thickness (mm)	.025	.101	.356	3.175	.635	3.175	3.175
Tensile modulus (M Pa)	. .	17.7	37.4	23.5	74.8	680	680
Thermal expansion (%/° C x 10^3)	. .	4.32	2.70	6.75	3.60	1.55	1.55
Life (yrs)	15 (tested)	5 (5% loss)	4	. .	Indef. (93° C causes 10% loss)	Indef.	Indef.

** Data not available

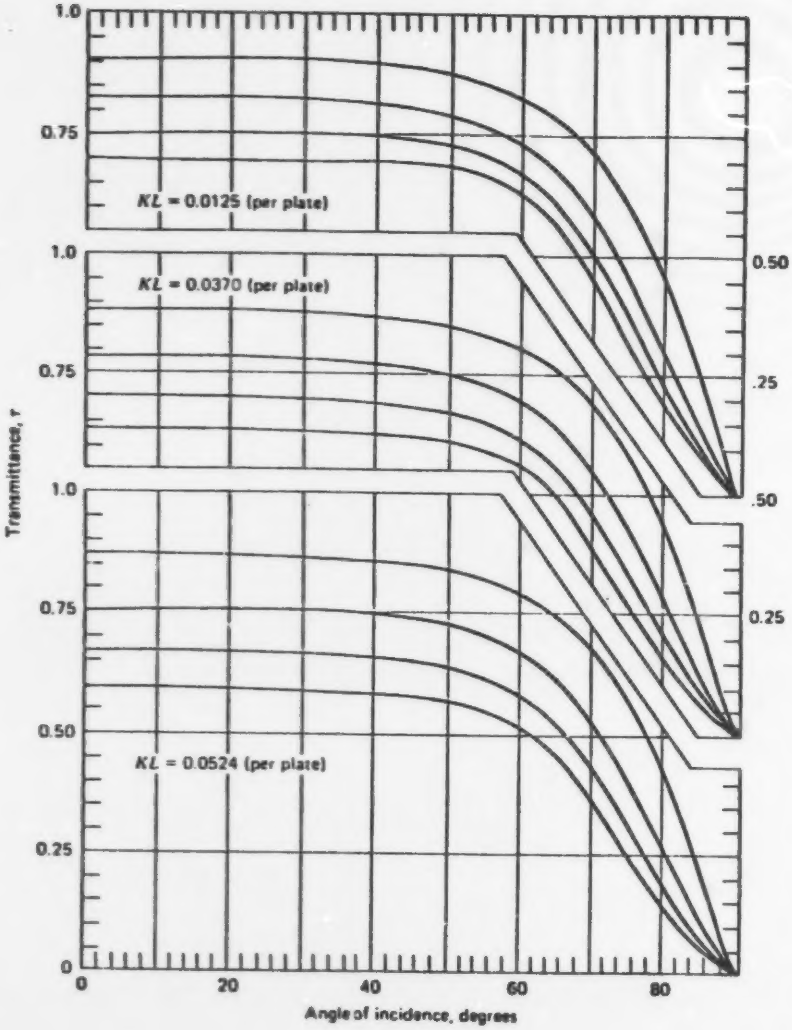


FIGURE 3.2.4

TRANSMITTANCE (CONSIDERING ABSORPTION AND REFLECTION)

OF 1,2,3 AND 4 COVERS FOR THREE TYPES OF GLASS

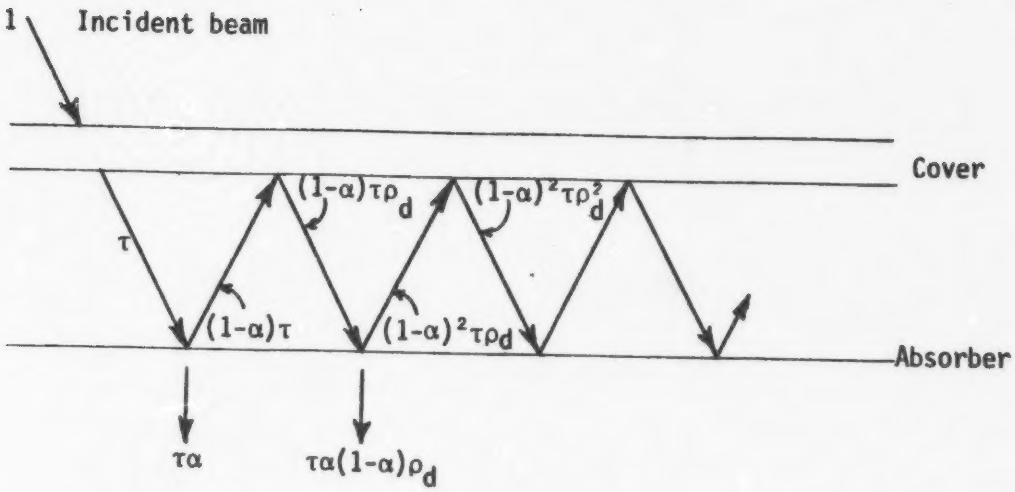

1
Incident beam
Cover
τ
$(1-\alpha)\tau\rho_d$
$(1-\alpha)^2\tau\rho_d^2$
$(1-\alpha)\tau$
$(1-\alpha)^2\tau\rho_d$
Absorber
$\tau\alpha$
$\tau\alpha(1-\alpha)\rho_d$

FIGURE 3.2.5
INTERACTION BETWEEN
COVER AND ABSORBER

Summing the absorbtion terms, the actual fraction of the incident beam absorbed is

$$(\alpha\tau) = \alpha\tau \sum_{j=o}^{\infty} \left[(1-\alpha)\ \rho_d\right]^j$$

$$= \frac{\alpha\tau}{1-(1-\alpha)\ \rho_d} \qquad \text{(Eqn 3.2.8)}$$

where ρ_d is the reflectance of the cover plates for diffuse radiation from below, and is approximately equal to ρ (60°).

The absorbtances of several surface coatings are given in Table 3.2.2.

Example 3.1

Find $(\alpha\tau)$ for two plates of the high quality glass from the previous two calculations over an absorber painted with 3M Black Velvet paint.

Table 3.2.2

ABSORBER SURFACE MATERIAL COMPARISON

Material	Solar Absorptance	Infrared Emittance	Absorptance Divided By Emittance
3M Black Velvet	.98	.90	1.1
Cat-a-loc flat black	.97	.86	1.1
Sicon flat black	.96	.83	1.2
Parson's black	.98	.92	1.1
Black Chrome Duramir BK	.97	.73	1.3
Nickel black on polished steel	.92	.11	8.4
Nickel black on galvanized iron	.89	.12	7.4
Nickel black, 2 layers on electroplated nickel on steel	.94	.07	13.0
CuO on nickel	.81	.17	4.8
Co_3O_4 on silver	.90	.27	3.3
CuO on aluminum	.93	.11	8.5
Cu Black on copper	.89	.17	5.2
Ebanol C on copper	.90	.16	5.6
CuO on anodized aluminum	.85	.11	7.7
PbS crystals on aluminum	.89	.20	4.5

Solution

For two cover plates

$$\tau_{r,2} = \frac{1-r}{1+(2N-1)r}$$

$$= \frac{1-.0434}{1+(4-1).0434}$$

$$= .846$$

$$\tau = \tau_{r,2}\ \tau_{\alpha}^{\ 2}$$

$$= .846 \times (.987)^2$$

$$= .825$$

$\alpha = 0.98$ (from Table 3.2.2)

$$\rho_d \cong \rho\ (60°)$$

and $\rho \cong \tau_{\alpha} - \tau$

From Figure 3.2.4, at KL = .0128 and at 60° incidence, τ=.73 for a two cover system.

$$\tau_{\alpha} = \varepsilon^{-KL} \qquad \text{(Eqn 3.2.6)}$$

$= .99$ Therefore $\rho_d = .99-.73$

$= .26$

$$(\alpha\tau) = \frac{\alpha\tau}{1-(1-\alpha)\ \rho_d} \qquad \text{(Eqn 3.2.8)}$$

$$= \frac{.98 \times .825}{1-(1-0.98).26}$$

$$= .813$$

3.3 HEAT TRANSFER IN FLAT PLATE COLLECTORS

3.3.1 Introduction

The useful energy collected by a flat plate collector is

$$Qu = A_c \left[S - U_L \left(T_{p,m} - Ta \right) \right] \qquad \text{(Eqn 3.3.1)}$$

where S = the energy absorbed/unit area

$= (\alpha\tau)\, I_T$

U_L = the overall heat loss coefficient

$T_{p,m}$ = the mean plate temperature

Ta = the ambient temperature

and A_c = the collector area.

The main problem of our analysis of collector heat transfer will be to modify this equation so that it can be written in terms of the known inlet temperature instead of the unknown mean plate temperature. Figure 3.3.1 shows the temperature distribution on the absorber plate. The temperature varies in a complex way both along and across the plate. From

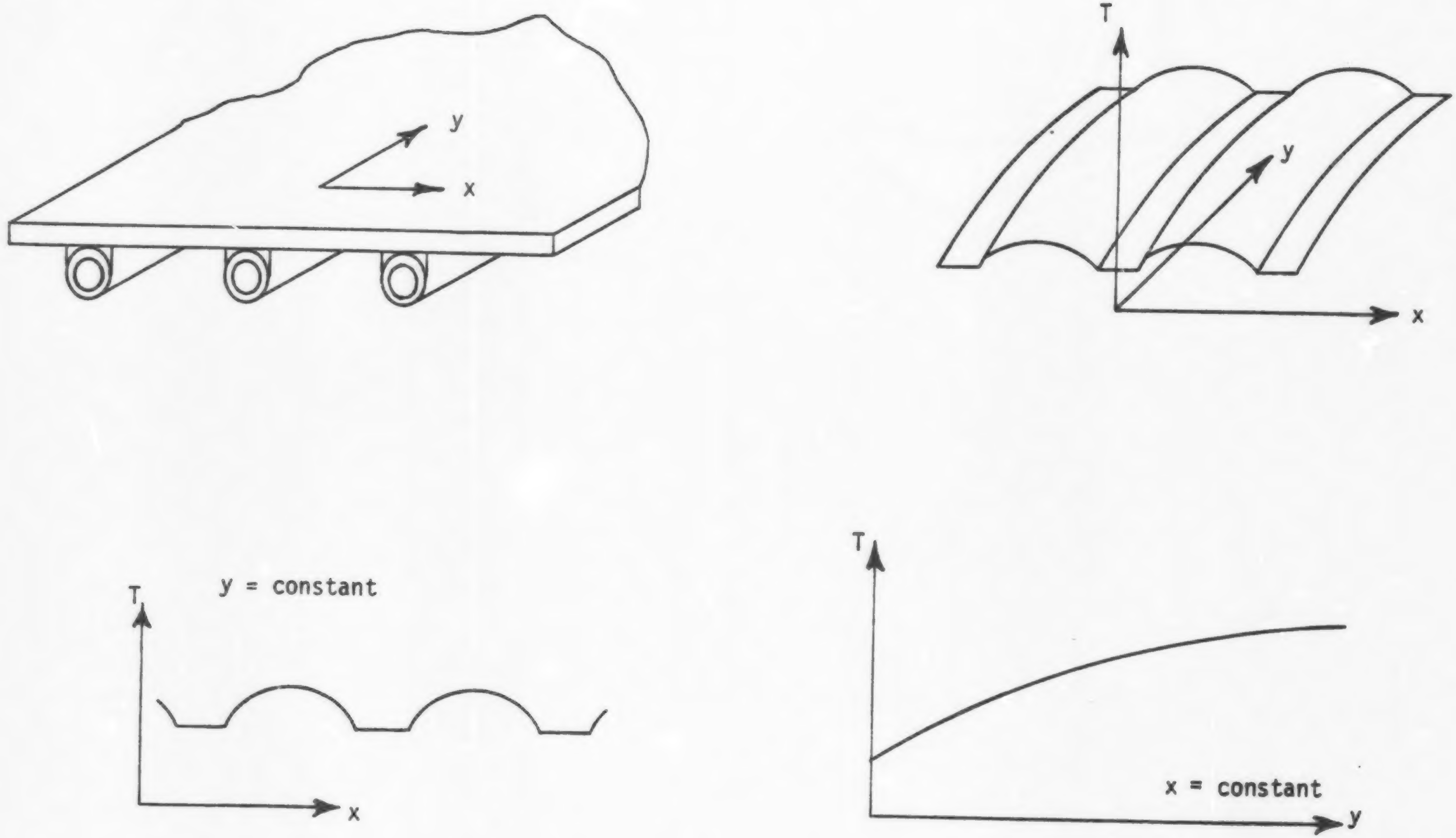
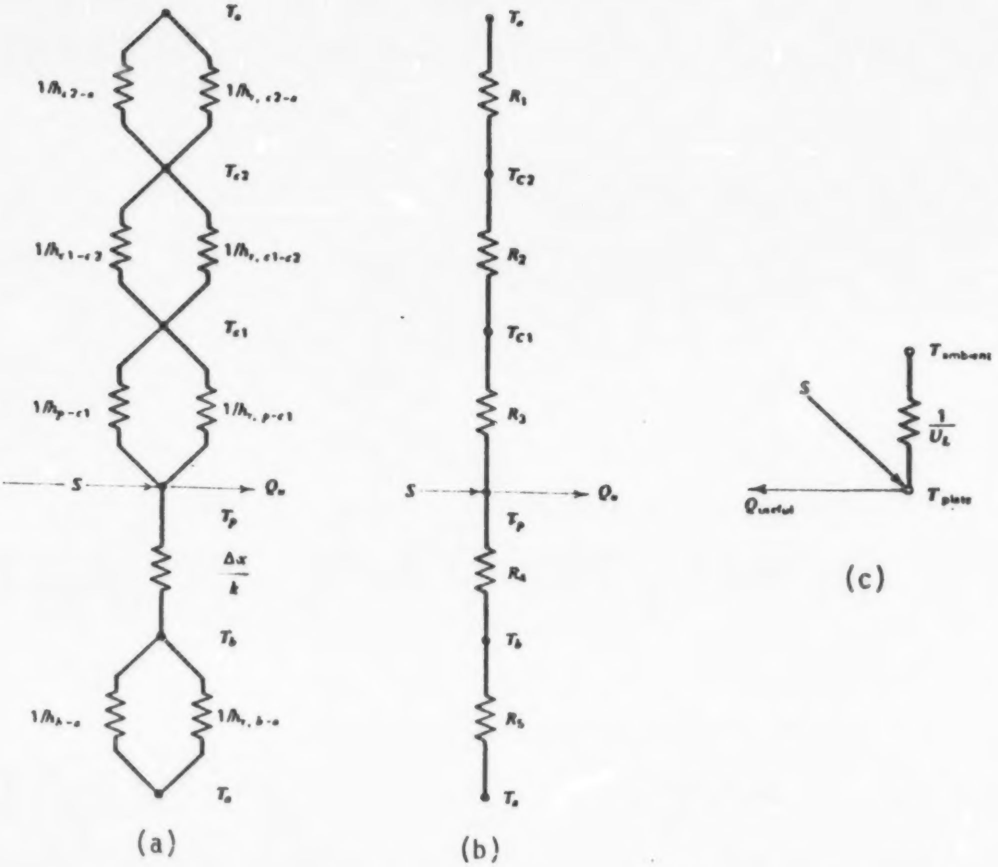

y
x
T
y
x
T
y = constant
x
T
x = constant
y

FIGURE 3.3.1

TEMPERATURE DISTRIBUTION ON ABSORBER PLATE


T_a
$1/h_{c2-a}$
$1/h_{r,c2-a}$
T_{c2}
$1/h_{c1-c2}$
$1/h_{r,c1-c2}$
T_{c1}
$1/h_{p-c1}$
$1/h_{r,p-c1}$
S
Q_u
T_p
$\frac{\Delta x}{k}$
T_b
$1/h_{b-a}$
$1/h_{r,b-a}$
T_a
T_a
R_1
T_{C2}
R_2
T_{C1}
R_3
S
Q_u
T_p
R_4
T_b
R_5
T_a
$T_{ambient}$
S
$\frac{1}{U_L}$
Q_{useful}
T_{plate}

(a)

(b)

(c)

FIGURE 3.3.2

THERMAL NETWORK FOR A TWO COVER

FLAT-PLATE COLLECTOR

(a) In terms of Conduction Convection, and Radiation resistances,

(b) In terms of resistances between plates, and

(c) In terms of overall loss coefficient this it can be seen that the mean plate temperature would be very difficult to measure or predict for a particular operating condition.

3.3.2 The Collector Heat Loss Coefficient

The radiant heat transfer between two parallel plates is given by

$$\frac{Q_r}{A} = \frac{\sigma\,(T_2^4 - T_1^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad \text{(Eqn 3.3.2)}$$

We can define a radiation heat transfer coefficient

$$h_r = \frac{\sigma(T_2 + T_1)\,(T_2^2 + T_1^2)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \qquad \text{(Eqn 3.3.3)}$$

so that $Q_r = h_r A\,(T_2 - T_1)$ (Eqn 3.3.4)

For radiant interchange with the sky

$$Q_r = \varepsilon A \sigma\,(T^4_{sky} - T^4) \qquad \text{(Eqn 3.3.5)}$$

With $h_r = \dfrac{\varepsilon\sigma(T_{sky} + T)\,(T^2_{sky} + T^2)(T - T_{sky})}{(T - T_{amb})}$ (Eqn 3.3.6)

we can write

$$Q_r = h_r\,A\,(T - T_{amb})$$

Note that these radiant heat transfer coefficients are not constants. They vary with the temperatures of the radiating bodies. Of course convective heat transfer coefficients also vary with temperature, but not nearly as much as radiative coefficients. Therefore, where radiation is an important mechanism of heat transfer, it is unlikely that heat flux will vary linearly with temperature difference over a wide range of temperatures.

The thermal resistances in a solar collector can be characterised as shown in Figure 3.3.2(a) and simplified as shown in Figures 3.3.2(b) and 3.3.2(c).

Thus $U_L = U_t + U_b$

where $U_t = \frac{1}{R_1 + R_2 + R_3}$ (Eqn 3.3.7)

and $U_b = \frac{1}{R_4 + R_5}$

with each of the thermal resistances which represents parallel radiation and convection given by

$$R = \frac{1}{h_r + h_c}$$

D and B present, on Pp 207-209, a series of graphs like those shown in Figure 3.3.3 for the top loss coefficients for various situations.

Figure 3.3.4 shows the magnitudes of the heat flows and cover plate temperatures for the following circumstances

$T_{plate} = 100°C$

$T_{amb} = T_{sky} = 10°C$

Plate spacing = 25 cm

Tilt = 45°

$h_{wind} = 10W/m^2\ °C$

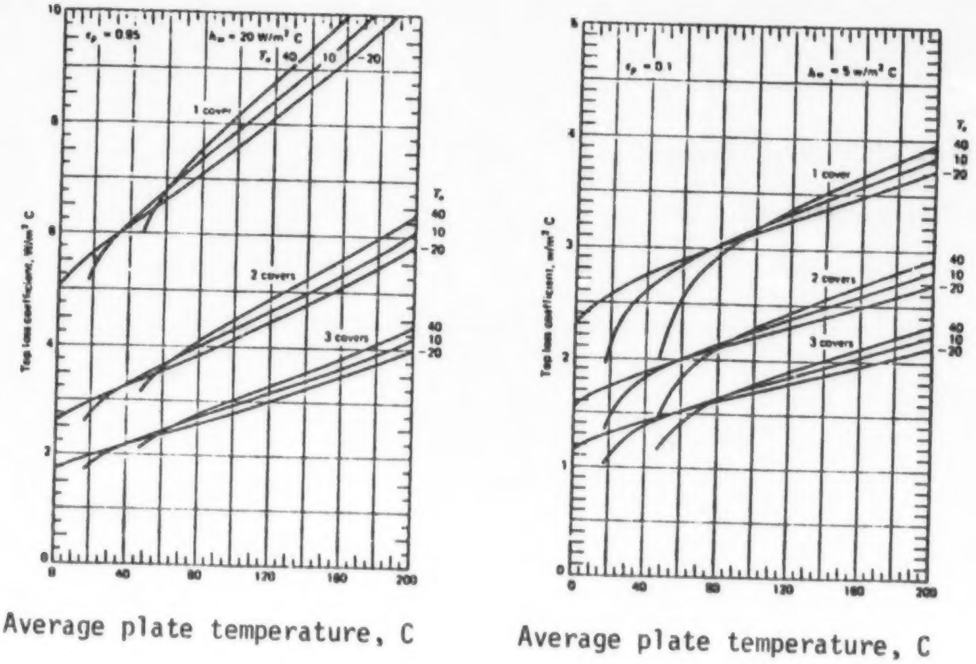


FIGURE 3.3.3

TOP LOSS COEFFICIENTS vs PLATE TEMPERATURE

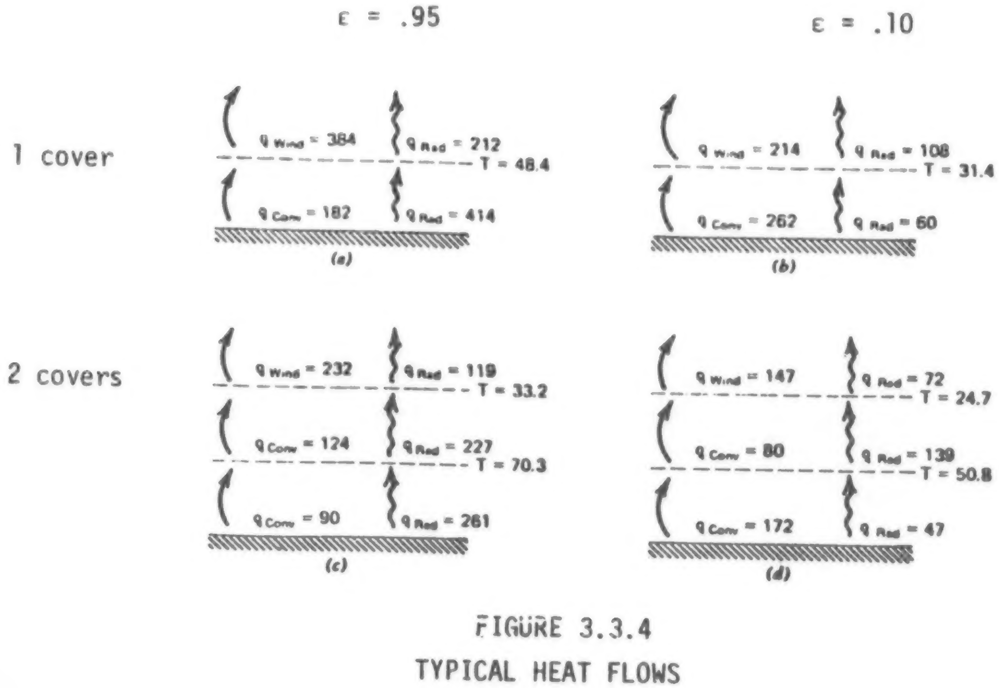

ε = .95
ε = .10
1 cover
q_{Wind} = 384
q_{Rad} = 212
T = 48.4
q_{Conv} = 182
q_{Rad} = 414
(a)
q_{Wind} = 214
q_{Rad} = 108
T = 31.4
q_{Conv} = 262
q_{Rad} = 60
(b)
2 covers
q_{Wind} = 232
q_{Rad} = 119
T = 33.2
q_{Conv} = 124
q_{Rad} = 227
T = 70.3
q_{Conv} = 90
q_{Rad} = 261
(c)
q_{Wind} = 147
q_{Rad} = 72
T = 24.7
q_{Conv} = 80
q_{Rad} = 139
T = 50.8
q_{Conv} = 172
q_{Rad} = 47
(d)

FIGURE 3.3.4

TYPICAL HEAT FLOWS

AND PLATE TEMPERATURES

Figure 3.3.5 shows the effects of plate spacing, absorber emittance and absorber temperature on U_L. It is interesting to note that absorber temperature has a substantial effect on U_L, even for a selective surface where radiation losses are minimized. Also notice that U_L is at a minimum at a cover plate spacing between 10 and 15 cm.

3.3.3 Temperature Distribution On The Plate Of A Conventional Liquid Cooled Collector

3.3.3.1 Temperature Distribution Across The Plate

The absorber plate is a fin transferring heat to the coolant tubes, and its thermal performance is like that of any other fin. The fin efficiency is defined as

F = (actual heat gain)/(heat gain if the whole fin were at the base temperature)

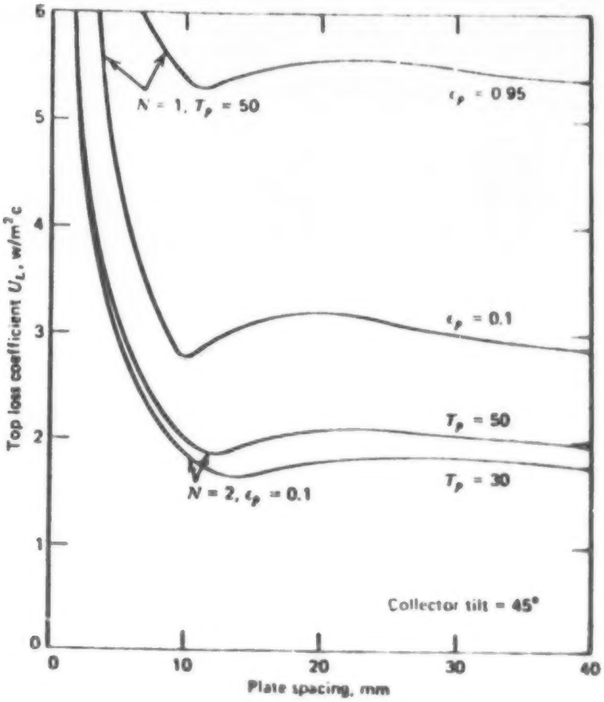


FIGURE 3.3.5
VARIATION OF TOP LOSS COEFFICIENT
WITH PLATE SPACING

F approaches 1.0 as the fin gets shorter or thicker or as the thermal conductivity approaches infinity.

Using the notation of Figure 3.3.6, the fin efficiency is given by

$$F = \frac{\tanh\,[m(W-D)/2]}{m(W-D)/2} \qquad \text{(Eqn 3.3.8)}$$

where $m = (U_L/k\delta)^{0.5}$

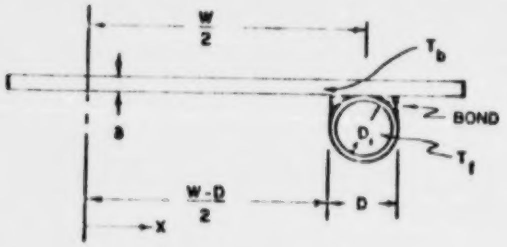


FIGURE 3.3.6
NOMENCLATURE FOR FIN ANALYSIS

Figure 3.3.7 shows the variation of F with m(W-D)/2.

The collector efficiency factor F' is defined as the actual useful energy gain divided by the energy gain if the absorber plate were at the fluid temperature. The collector efficiency factor is given by

$$F' = \frac{1/U_L}{W\left[\dfrac{1}{U_L[D+(W-D)F]} + \dfrac{1}{C_b} + \dfrac{1}{\pi D_i h_{f,i}}\right]} \qquad \text{(Eqn 3.3.9)}$$

C_b is the bonding coefficient between the absorber and the coolant tubes (in units of W/°C per meter of the tube length) so that $1/C_b$ is the plate to tube thermal resistance per unit length of tube. $h_{f,i}$

is the convective heat transfer coefficient inside the tubes, so that $\frac{1}{\pi D_i h_{f,i}}$ is the tube to fluid thermal resistance per unit length of tube. $[D+(W-D)F]$ is the tube pitch, corrected by the fin efficiency F for the fact that there is a thermal resistance between some parts of the plate and the place where they bond to the tubes. Thus $\frac{1}{U_L[D+(W-D)F]}$ is the thermal resistance between the environment and

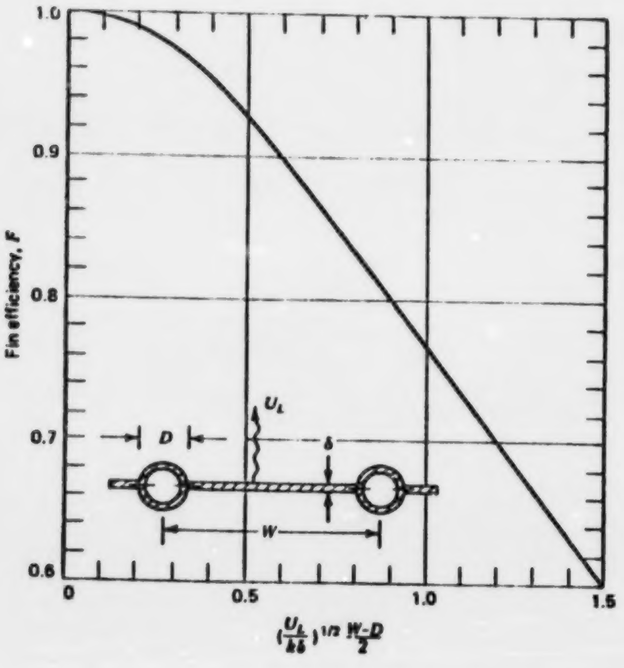


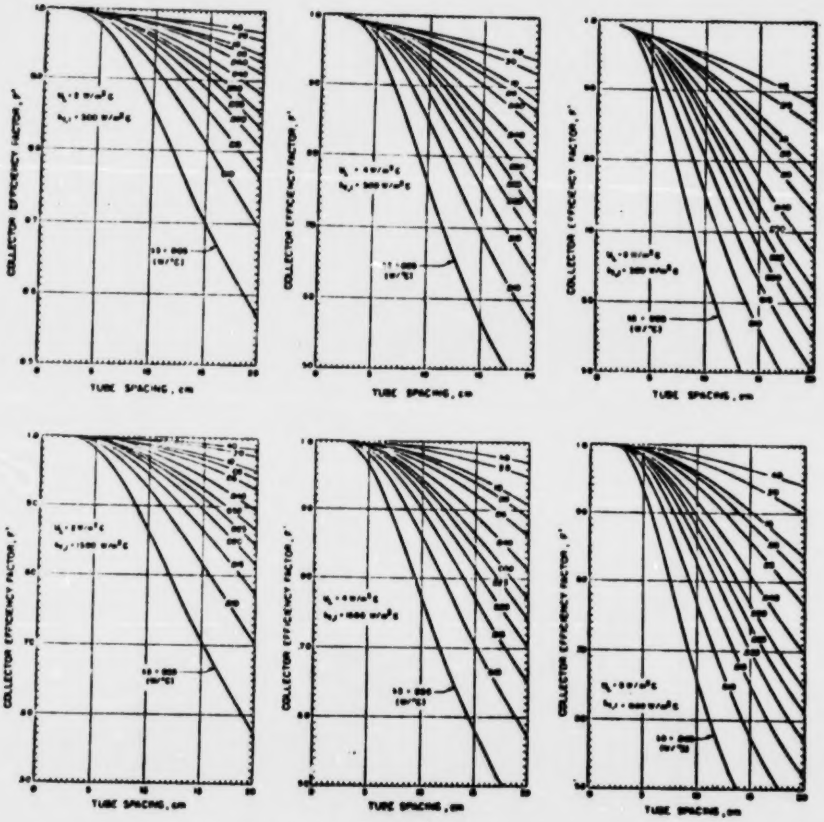
FIGURE 3.3.7

VARIATION OF FIN EFFICIENCY

WITH M(W-D)/2

the place where the plate bonds to the tubes (per unit length of tube) Thus $F' = R_L$ Ro where R_L is the thermal resistance from the plate to the environment and Ro is the thermal resistance from the tube to the environment.

Figure 3.3.8 shows the variation of F' with tube spacing, U_L, $h_{f,i}$ and kδ for 20 mm diameter tubes. Duffie and Beckman present similar data for 10 mm tubes in Figure 6.5.4, Pp 219-221.



Collector efficiency factor F' versus tube spacing for 2 cm diameter tubes for various conditions.

FIGURE 3.3.8

3.3.3.2 Temperature Distribution Along The Plate

The thermal performance of a solar collector is decreased not only because the plate temperature increases between coolant tubes, but also because the fluid temperature, and thus the plate temperature, increases from the inlet to the outlet. As the fluid temperature increases, the local efficiency and thus the rate of temperature increase, is lowered, resulting in a temperature profile like that shown in Figure 3.3.9.

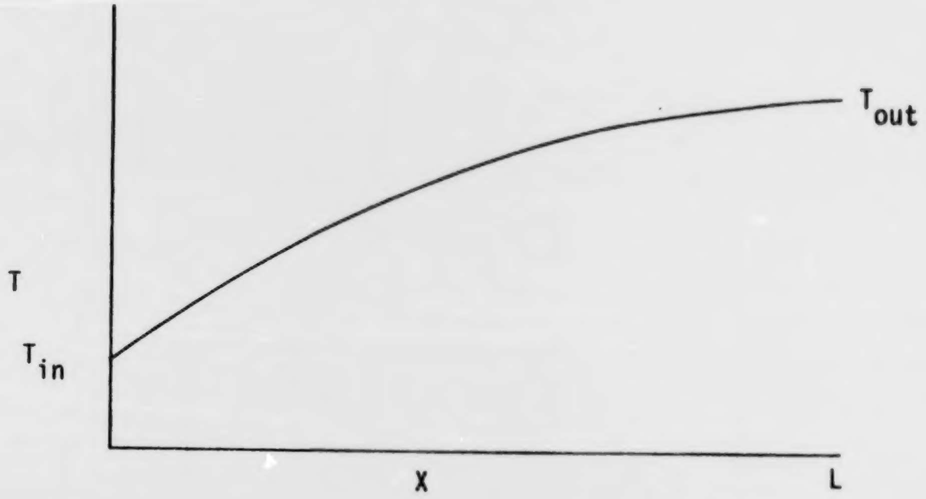


FIGURE 3.3.9

FLUID TEMPERATURE vs

POSITION ALONG TUBE

To account for this decrease in performance we introduce the flow factor

F" = (energy gain)/(energy gain if the fluid temperature were everywhere equal to the inlet temperature).

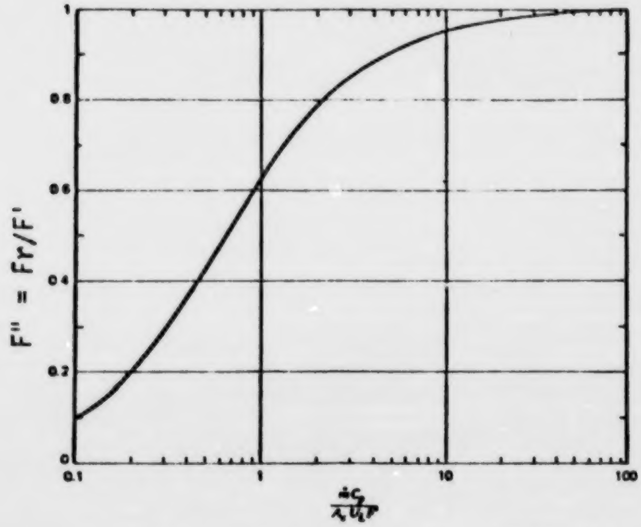
Obviously, if the fluid thermal capacity rate, $\dot{m}C_p$, were very large, F" would approach 1.0.

It can be shown (see D and B Sec 6.7) that

$$F'' = \frac{1}{z}\left(1 - e^{-z}\right) \qquad \text{(Eqn 3.3.10)}$$

where $z = \dfrac{A_c F' U_L}{\dot{m}\, C_p}$

Figure 3.3.10 shows F" as a function of 1/z.



Collector flow factor F'' as a function of $\dot{m}C_p/A_c U_L F'$.

FIGURE 3.3.10

Combining the two previous concepts, it becomes possible to introduce the collector heat removal factor

$$F_R = F'F'' \qquad \text{(Eqn 3.3.11)}$$

This is the ratio of the useful energy gain to the useful energy gain which would occur if the entire plate were at the fluid inlet temperature. Combining equations 3.3.10 and 3.3.11

$$F_R = \frac{\dot{m}\,C_p}{A_c U_L}\left(1 - e^{-A_c U_L F'/\dot{m}\,C_p}\right) \qquad \text{(Eqn 3.3.12)}$$

Introducing F_R into Eqn 3.3.1 yields

$$Q_u = A_C F_R \left[S - U_L (T_{f,i} - T_a)\right] \qquad \text{(Eqn 3.3.13)}$$

where $S = G_T (\alpha\tau)$

However, there is a further correction to make to this equation. When $(\alpha\tau)$ was evaluated, the effect of the radiation absorbed by the cover plates on their temperature was ignored. This slight increase in temperature will decrease the losses from the absorber. This can be taken into account by introducing the effective transmittance-absorbtance product,

$$(\alpha\tau)_e = (\alpha\tau) + \left(1 - \tau_{a,1}\right) a_1$$
$$+ \left(1 - \tau_{a,2}\right) a_2 \tau_1 + \left(1 - \tau_{a,3}\right) a_3 \tau_2 + \cdots \qquad \text{(Eqn 3.3.13)}$$

where a_i is the ratio of the top loss coefficient to the loss coefficient from the i_{th} cover to the atmosphere. For example, for a single cover of low iron glass over a black painted surface

$$(\alpha\tau)_e = (\alpha\tau) + \left(1 - \tau_a\right) a_1$$
$$= (.98 \times .906) + (1 - .988)\ .427$$
$$= .888 + .005$$
$$= .893$$

In this case, the correction is of the same magnitude as the errors in the property data. Only in double glazed collectors with cover plates having high absorbtion does this correction become significant.

Equation 3.3.13 can now be written as

$$Q_u - A_C F_R G_G (\alpha\tau)_e - A_C F_R U_L (T_{f,i} - T_a) \qquad \text{(Eqn 3.3.14)}$$

3.3.4 Other Collector Geometries

The analysis of Section 3.3.3 applies to a liquid cooled tube in plate type collector of conventional design. A similar approach has been taken for many other geometries. It is usually possible to express the results in a similar form, with slightly different expressions for F' and/or F_r. D and B present the results of several such analyses in Section 6.12, beginning on Pg. 236.

A geometry which is used in one of the PUSH approved liquid cooled collectors is the serpentine tube. D and B, in Example 6.12.1 presents an analysis of F_r for a serpentine tube collector. A parallel tube collector of the same size, materials and mass flow rate would have about the same value of F_r, but the pressure drop to maintain this flow rate would be considerably less.

In general, air cooled collectors will have lower values of F_r than liquid cooled collectors, because of the lower specific heat, lower mass flow rate and lower heat transfer coefficient. However, an air cooled collector using a porous absorber plate through which the air is drawn can have a value of F_r very close to 1.0. Such a collector is shown in Figure 6.12. i, Pg. 239, D and B. One of the PUSH approved air cooled collectors uses this geometry.

3.4 FLAT PLATE COLLECTOR THERMAL PERFORMANCE

ASHRAE Standard 93-77 prescribes the standard method of test for rating the thermal performance of flat plate collectors. (Note that this standard is presently undergoing revision and a new version will be published in 1983.) This standard test method is based on the heat transfer analysis of Section 3.3.

The efficiency of a flat plate collector is defined as

$$\eta = Q_u \ A_c G_t \qquad \text{(Eqn 3.4.1)}$$

Substituting Q from equation 3.3.14

$$\eta = F_R(\alpha\tau)_e - F_R U_L \left(\frac{T_{f,i} - T_a}{G_t} \right)$$

$F_R(\alpha\tau)_e$ and $F_R U_L$ are properties of the collector. They vary only slightly with temperature. $\left(T_{f,i} - T_a\right)/G_t$ depends only on conditions external to the collector. Therefore, for any one collector, the effiency can be plotted in the form $\eta = a + bx$ with the y-intercept $a = F_R(\alpha\tau)_e$ and the slope $b = F_R U_L$.

The abscissa $(T_{f,i} - T_a)/G_t$ combines all the external variables into a single term.

Figure 3.4.1 shows a typical ASHRAE 93-77 efficiency plot, prepared by fitting a straight line through four sets of four data points, measured over a wide range of values of $\left(T_{f,i} - T_a\right)/G_t$. It is important to note that the efficiency curve is based on gross area, so that losses due to radiation falling on the collector frame instead of the cover plate lower the apparent collector efficiency. The ASHRAE 93-77 collector thermal performance tests are carried out with radiation falling normally on the collector, so the "incident angle modifier" must also be measured, so that the higher reflective losses at incident angles away from normal can be accounted for. Figure 3.4.2 shows a typical incident angle modifier curve.

The Incident Angle Modifier is defined as:

$$K_{\alpha\tau} = (\alpha\tau) / (\alpha\tau)$$

so that the useful energy collected is then given by

$$Qu = A_c F_R \left[G_T K_{\alpha\tau} (\alpha\tau)_n - U_L (T_i - T_a)\right]$$

The variation of $K_{\alpha\tau}$ with incident angle, θ, can be approximated by the expression

$$K_{\alpha\tau} = 1 + b_o (1/\mathrm{Cos}\ \theta - 1)$$

where b_o is a constant, called the incident angle modifier coefficient, which can be determined experimentally for each collector.

The Canadian Solar Industries Association publishes descriptions of all PUSH approved solar collectors. On the last page of each collector description are the results of an ASHRAE 93-77 test of the collector. The results shown are only valid for the same heat transfer fluid at the same flow rate. UNIES Ltd. has prepared a set of graphs for the Solar Programs Office which will be published shortly and which will provide correction factors for the common fluids over a range of flow rates.

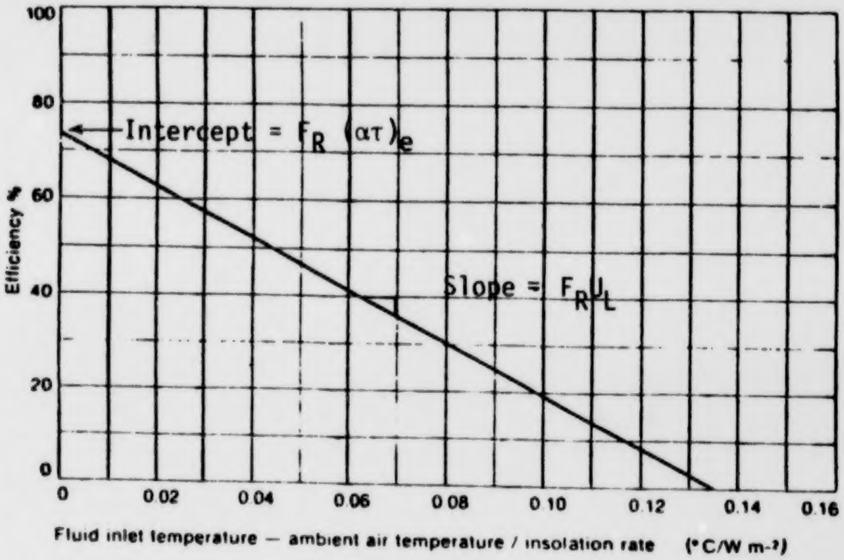


FIGURE 3.4.1

COLLECTOR EFFICIENCY CURVE

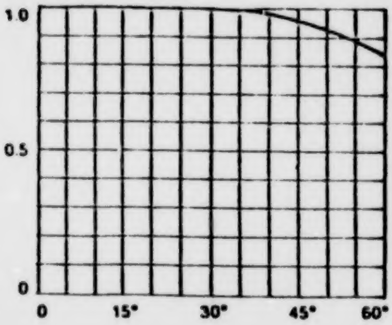


FIGURE 3.4.2

INCIDENT ANGLE MODIFIER

Example

A solar collector has a copper absorber 0.5 mm thick, coated with 3M Black Velvet paint. It is double glazed with water white (low iron) glass. The coolant tubes are 2 cm. in diameter and are spaced at 15 cm. intervals. The collector is mounted at 45°. The wind heat transfer coefficient is 10 W/m^2 °C. The ambient temperature is 10°C, the fluid inlet temperature is 40°C, the coolant is water flowing at 3.5ℓ/min. The heat transfer coefficient inside the tubes is 1500W/m^2 °C, and the collector area is 1.7 m^2. The back loss coefficient is 1.0 W/m^2 °C. Find the expected values of $F_R(\alpha\tau)$ and F_RU_L.

Solution

From previous examples, for this absorber-cover combination $(\alpha\tau) = .0.82$.

For copper, $K = 386$ W/m °C.˙. $K\delta = 386 \times .0005 = 0.193$ W/°C.

From Figure 6.4.4(b), Pg 207, D and B,$U_{TOP} = 3\text{W/m}^2$ °C.

Therefore $U_L = U_{TOP} + U_{BACK}$

$$= 3 + 1$$

$$= 4.0 \text{ W/M}^2 \text{ °C}$$

$$\text{and}\left(\frac{U_L}{K\delta}\right)^{\frac{1}{2}} \frac{W-D}{2} = \left(\frac{4}{.193}\right)^{\frac{1}{2}} \times \frac{.13}{2}$$

$$= .2959$$

From Figure 3.3.7, F = .9718

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L(D+(W-D)F} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{f,i}}\right]}$$

$$= \frac{1/4}{.15\left[\frac{1}{4\ (.02 + .13 \times .9718)} + 0 + \frac{1}{\Pi \times .02 \times 1500}\right]}$$

$$= \frac{.25}{.15\ [1.7084 + .0106]}$$

$$= .9695$$

$$\therefore \left(\dot{m}\ C_p/U_L F^1 A_c\right) = \frac{3500/60 \times 4.184}{4.0 \times .9695 \times 1.7}$$

$$= 37.02$$

From Figure 3.3.9, F" = .9866

$\therefore F_R = F'F" = .9695 \times .9866$

$= .957$

$\therefore F_R(\alpha\tau) = .957 \times .83$

$= .79$

and $F_R U_L = .957 \times 4.0$

$= 3.9 \text{ W/m}^2 \text{ °C}$

Assignment

A solar collector has an absorber plate of steel 1 mm thick. The abosrbing surface is nickel black. The tube spacing is 15 cm and the tube diameter is 2 cm. The collector has a single cover plate of .014" thick mylar. The mass flow rate of the cooling water is 2 ℓ/min and the heat transfer coefficient inside the tubes is 300 W/M² °C. The collector area is 1.7 m². Find F_R ($\alpha\tau$) and $F_R U_L$.

3.5 COLLECTOR ARRAY PERFORMANCE

All the above analysis applies to individual collectors. When the collectors in an array are connected in parallel, each collector receives coolant at the same inlet temperature, so the same analysis is valid. When collectors are connected in series, so that the coolant which has been heated in one collector enters the next, this is not the case. Therefore the results obtained above must be modified to take this into account.

One PUSH approved collector manufacturer, Nortec Solar Industries Inc., provides a graph in their product information brochure which allows the collector performance to be corrected for series mounting. Where this information is not available, it is possible to correct the collector performance by assuming that two collectors in series will perform the same as a single collector with half the flow. Methods of carrying out this correction will be discussed in Part 2 of this course.

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CHAPTER 4

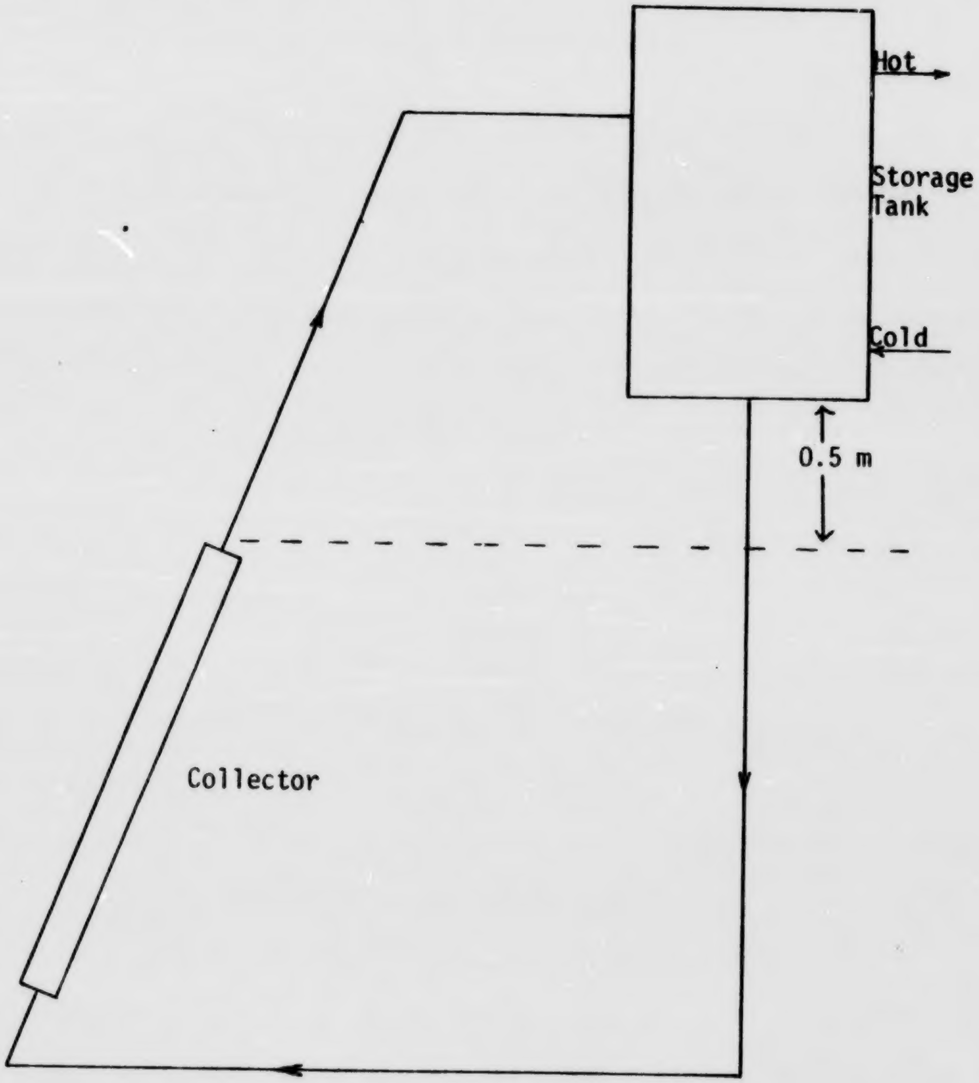
LIQUID SYSTEMS

The design of a solar energy system is dependent on use of the energy collected and on the physical and environmental conditions the systems must operate in. The most common uses made of solar energy are space heating, service water heating and swimming pool heaters. Solar cooling systems using absorption refrigerating cycles are commercially available, but due to their poor economics, are not discussed here.

4.1 TYPES OF SYSTEMS

The simplest use of solar energy is service water heating using a direct heating passive design. This type of system ranges from outdoor metal water storage tanks found throughout the tropics which may or may not be glazed to reduce thermal losses, to thermosiphon circulation systems. These systems are for non-freeze installations, so their use in Canada is limited to summer operation in all localities.

The thermosiphon circulation system (Figure 4.1.1) may operate in a pressure system, or in an open system. Solar radiation warms the water in the collectors. This water rises through the collector by convection into the top of the storage tank, which is located above the top of the collector. Cold water from the bottom of the storage tank is piped into the bottom of the collectors, displacing the warm outlet water. The bottom of the storage tank must be 0.5 m or more above the top of the collectors. This elevation difference, along with short, well insulated pipe runs en-


Hot
Storage Tank
Cold
0.5 m
Collector

Direct Heating Thermosiphon Circulation Type of Solar Waste Heater

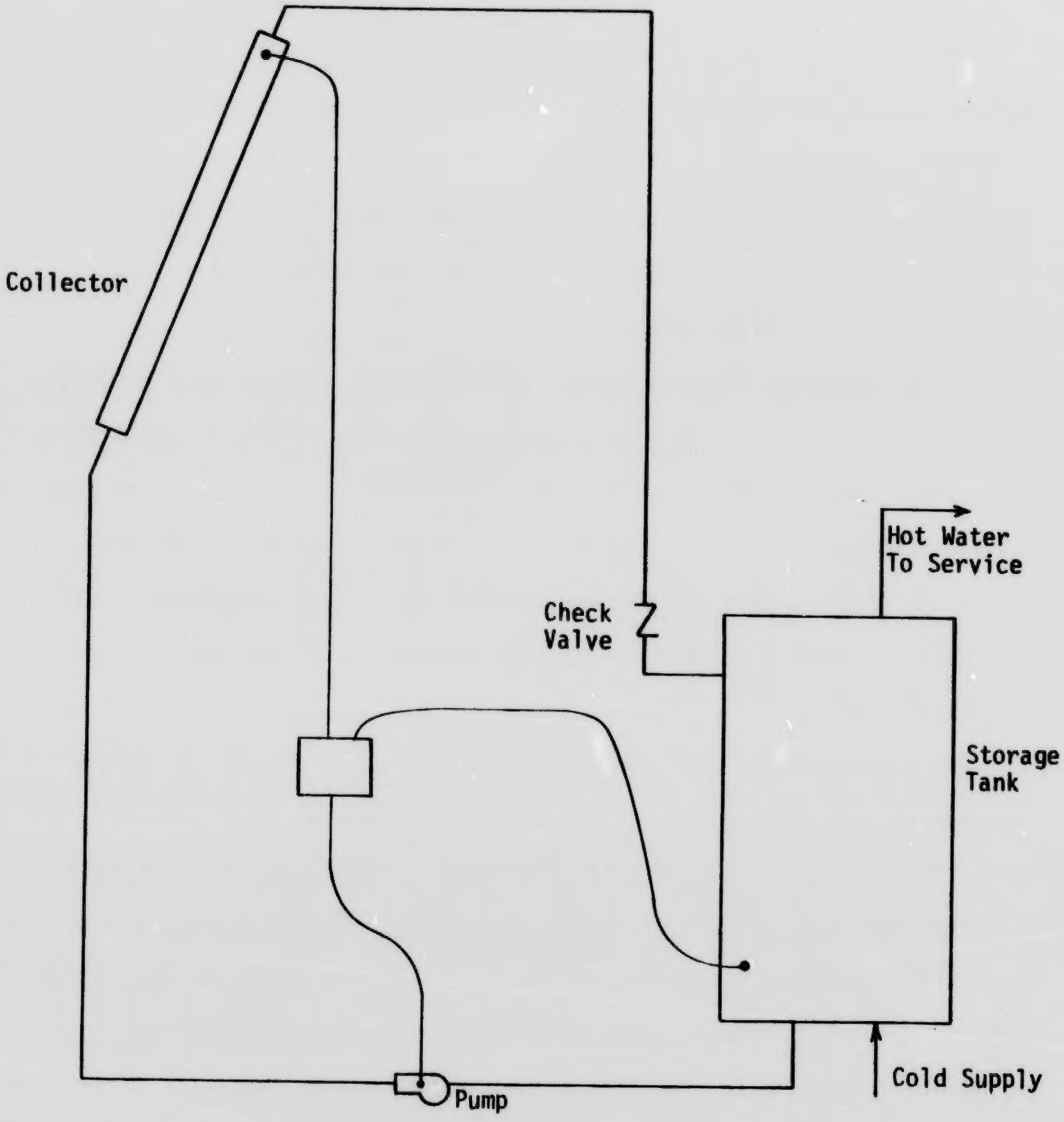
FIGURE 4.1.1

courages convective movement which will limit the temperature rise in the collectors, thus improving efficiency. If the solar collectors are not located completely below the storage tank, when the collector temperature falls below that of the storage, convection currents may reverse, and major heat losses can be incurred.

A high degree of thermal stratification in the storage tank results and so the collectors utilize only the coldest water from storage, with the highest temperature water being available for service. As solar radiation increases, so does the rate of circulation. ASHRAE comparison tests show thermosiphon systems are more efficient than mechanically driven circulation systems, largely due to required deadbands and inaccuracies in control settings.

The major problem with thermosiphon circulation type water heaters is the critical relationship of the relative locations of the storage tank and collectors. The simplest way to circumvent this is to install a circulation pump (Figure 4.1.2) in the circulation loop. Controls to operate the pump can consist simply of a temperature sensor in the collectors but for more efficient operation, a temperature comparator should be used with sensors in the storage tank and on the collectors. If the collectors are located above the storage tank, a check valve is required to prevent heat losses caused by the reverse thermosiphoning of stored water through the collectors when the pump is off.

These are direct heating systems. That is, water is heated directly in the collectors. Direct systems such as these require some form of


Collector
Hot Water
To Service
Check
Valve
Storage
Tank
Cold Supply
Pump

Direct Heating, Pump Circulation Type Of Solar Water Heater

FIGURE 4.1.2

freeze protection in Canada. Drain back and drain-down systems can provide this protection by automatically draining all the water from the exposed system components when freezing temperatures are approached or when the system is not in operation. Fail safe designs are critical to long term system operation.

A drainback system (Figure 4.1.3) usually operates with an unpressurized thermal storage system. The solar fluid circulation lines are vented and graded so that the solar fluid will drain back to the thermal storage tank when the circulation pump is not operating. When the circulation pump starts, the system will be re-filled with solar fluid.

Advantages of drainback systems include:

(1) system has freeze protection

(2) controls are simple

(3) they can use water as the solar fluid. Water is cheap and doesn't degrade with use.

(4) being direct heating systems, they enjoy high operating efficiencies

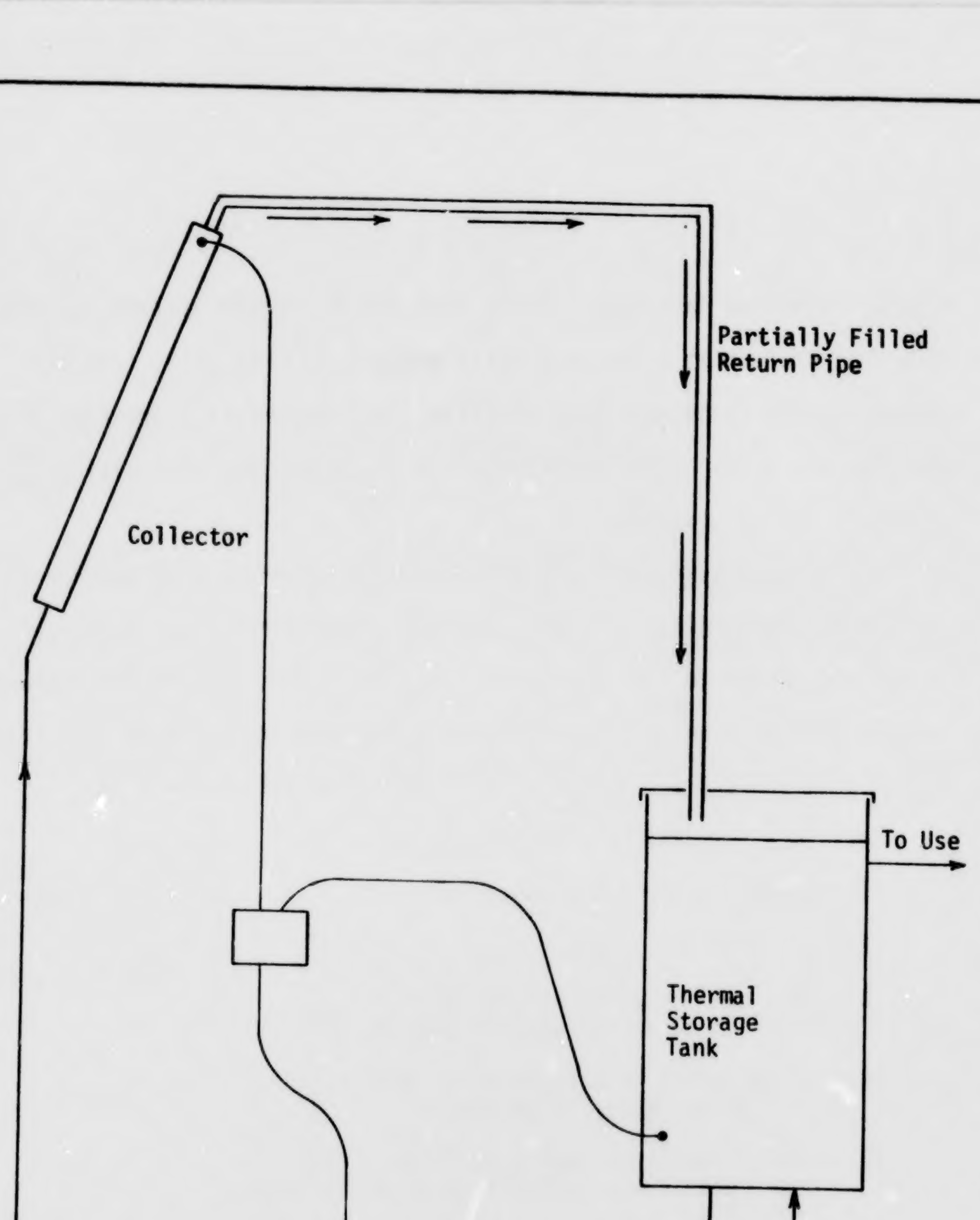
(5) the system is mechanically simpler and thus cheaper than other systems with freeze protection

(6) drainage of solar fluid from collectors to thermal storage results in recovery of solar energy which might otherwise be lost to the atmosphere when the system shuts down.

The disadvantages of drainback systems are:

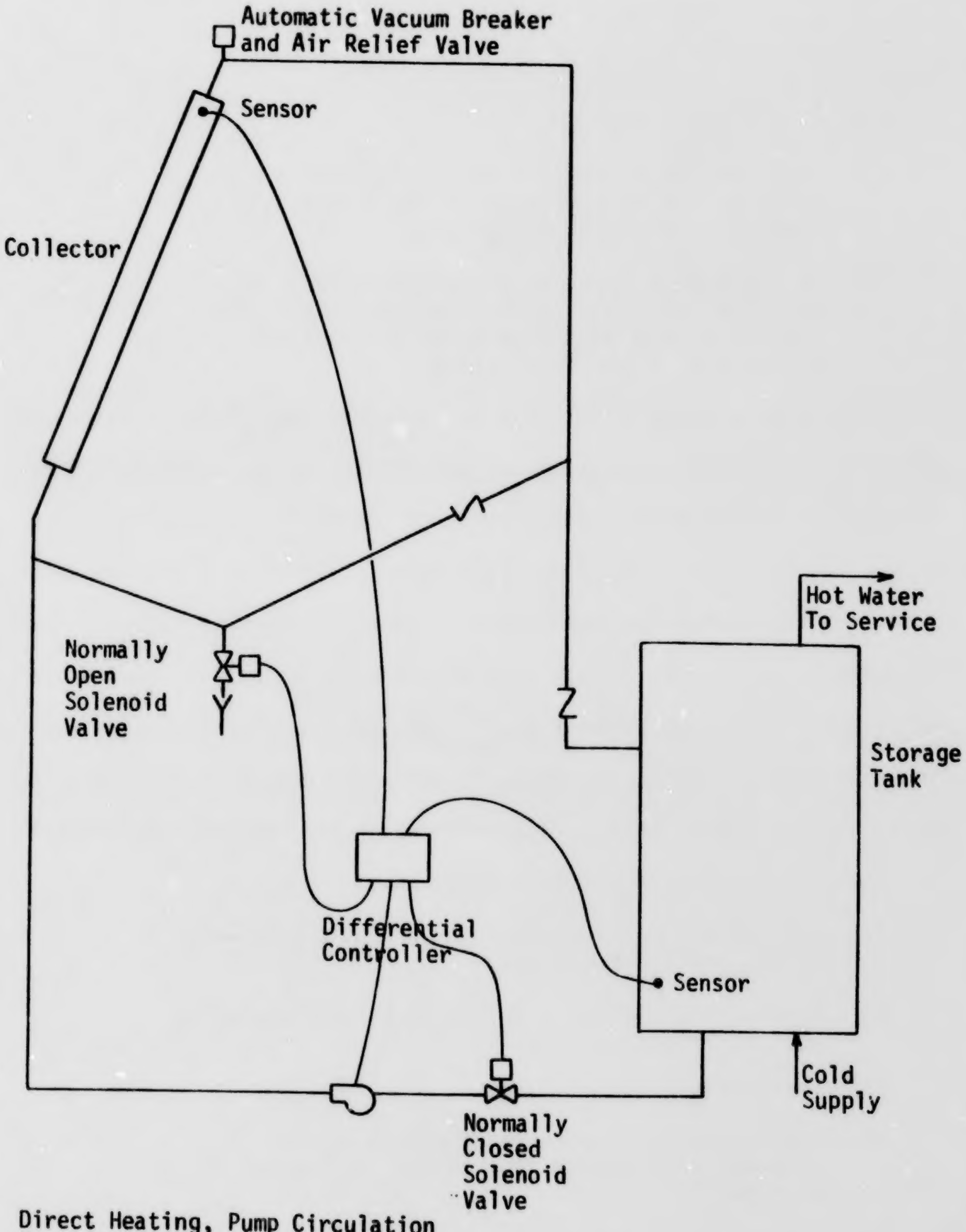
(1) frequent exposure of the wet piping and collector surfaces to air accelerates corrosion.

(2) all components and piping must be graded to permit complete drainage of outdoor components.


Partially Filled
Return Pipe
Collector
To Use
Thermal
Storage
Tank
Return

Gravity Return Drain Back
Solar Water Heater

FIGURE 4.1.3


Automatic Vacuum Breaker
and Air Relief Valve
Sensor
Collector
Hot Water
To Service
Normally
Open
Solenoid
Valve
Storage
Tank
Differential
Controller
Sensor
Cold
Supply
Normally
Closed
Solenoid
Valve

Direct Heating, Pump Circulation Type of Solar Water Heater with Automatic Drain Down

FIGURE 4.1.4

(3) the circulation pump has to pump against a head equal to the highest point in the system, thus increasing the pumping requirements.

(4) if the thermal storage is not pressurized, a pressure system is required to integrate a domestic hot water system or other pressurized system into a drainback system.

A draindown system (Figure 4.1.4) overcomes some of the previous objections by operating a closed loop, pressurized solar energy system having controls to drain the system down when the differential temperature between the collectors and the storage tank is less than a present value or cold ambient conditions are sensed. Check and solenoid valves isolate the pressurized thermal storage tank from the solar loop and then drain the solar loop to a waste drain when signalled. Once satisfactory running conditions re-occur, automatic valves would revert to the operating positions, the system would re-fill with water and the pump would start.

The advantages of draindown systems are:

(1) This is a direct heating solar energy system with freeze protection.

(2) Pumping requirements are less than those of drainback systems.

(3) The system is pressurized.

(4) The frequency of exposure of piping to the atmosphere is reduced from that of drainback systems.

The disadvantages of a draindown system are:

(1) during transient solar conditions in cold weather, the system may cycle frequently, wasting water which has already been heated. This increases the operating costs (purchase of metered water) and decreases the system efficiency.

(2) as with the draindown system, piping must be properly graded to permit complete drainage.

(3) proper control operation is critical. Control failures may result in freeze damage to the system or may result in large volume water spills.

(4) automatic air vents and vacuum breakers may freeze after the system is drained and cooled, resulting in a water spill at the collector array.

For these reasons, drain down systems are most suitable in locations where the occurrence of freezing temperatures is rare. Most continuous use liquid solar energy systems designed for Canadian installations are Circulating Type, (Figure 4.1.5) commonly called dual fluid systems. These systems circulate the transfer medium, which is a non-freezing liquid, through the collectors where it absorbs solar energy before passing through a heat exchanger where its heat is transferred to a storage medium, usually water. The transfer media are not generally used for thermal storage because their heat storage capacities are less than that of water, and they usually are relatively expensive. The transfer medium is most commonly ethylene glycol and water or propylene glycol and water. Dual fluid systems overcome freezing problems at the expense of additional equipment, complexity and a slight decrease in system performance. Other disadvantages of dual fluid systems are related to the degradation of solar fluid and the necessity to prevent solar fluid from entering domestic water supplies.

Outdoor swimming pools present a very good opportunity for solar heating. The use period corresponds to that part of the year with high solar availability while frequently ambient temperatures will exceed pool

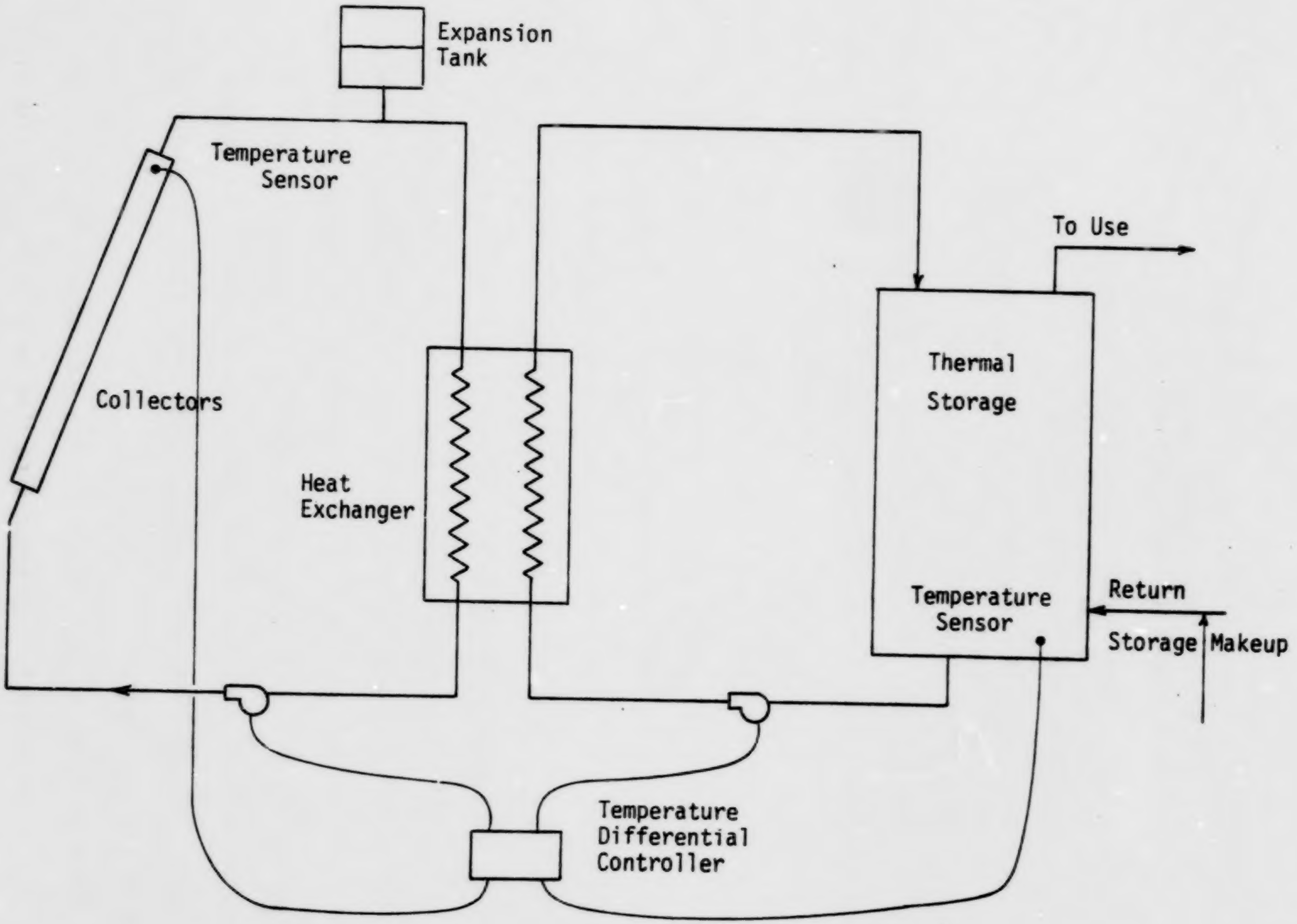

Expansion Tank
Temperature Sensor
Collectors
Heat Exchanger
To Use
Thermal Storage
Temperature Sensor
Return
Storage Makeup
Temperature Differential Controller

FIGURE 4.1.5
DUAL FLUID SYSTEM

temperatures. This means that the solar collector may gain energy from the ambient air as well as from solar radiation. Swimming pool systems are generally integrated into the pools filtering and auxilary heating circuits. The pool itself serves as thermal storage. The factors which make solar heating of swimming pools one of the most economical applications for solar energy include:

(1) The normal relationship between collector operating temperature and ambient air temperatures allows low cost, uninsulated collectors to achieve high operating efficiencies.

(2) Low operating pressures allow the use of low cost materials for collector absorber plates(such as extruded plastic sheets).

(3) Swimming pools have a large thermal mass which precludes the need for additional thermal storage.

(4) Summer only operation means the collector array is not very sensitive to slope or azimuth variation. This insensitivity frequently allows the engineer to design the system into existing site features, which will greatly reduce first costs.

(5) Summer only operation eliminates the need for freeze protection. Plastic collectors, which are frequently used in pool systems, can generally tolerate several freeze-thaw cycles without experiencing significant structural damage.

(6) Swimming pools have a relatively large heating load which occurs during that part of the year with a high availability of solar radiation.

(7) The simplicity of most swimming pool solar heating systems makes them technically and economically acceptable.

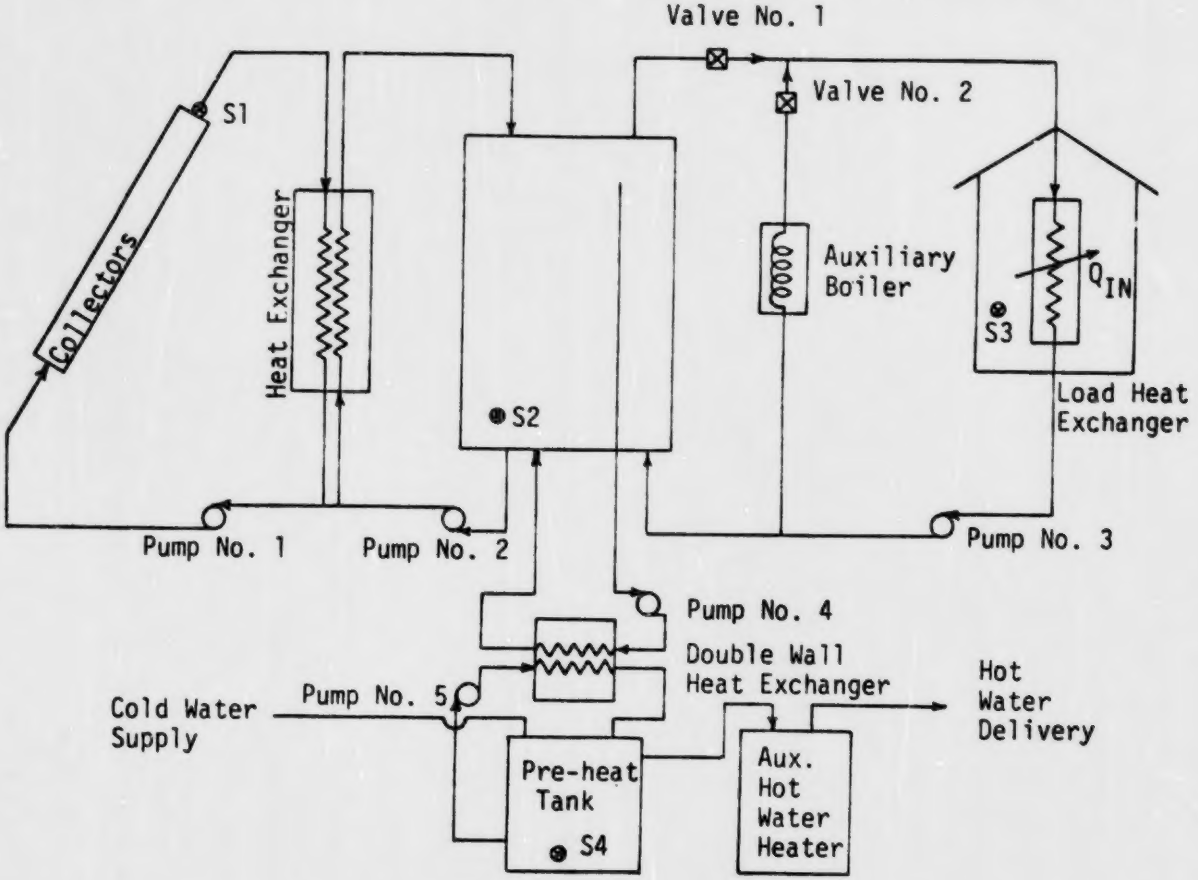
4.2 GENERAL DESIGN CONSIDERATIONS

4.2.1 Stored Heat To Use

The utilization of the collected energy determined its method of extraction from thermal storage. In a domestic water heating system, preheated water may be taken directly from thermal storage into the water system. In combined DHW - space heating sytems, the thermal storage may be contaminated by contact with heating system components, precluding direct use of storage water in the DHW system. A second heat exchanger is shown in Figure 4.2.1 is used to preheat the DHW in these systems.

4.2.2 High Temperature Protection

In solar DHW systems, there is a possibility that during times of low water use, DHW temperatures will exceed safe limits. The installation of a mixing valve downstream of the DHW heater will temper the water stream to the recommended temperature. DHW systems with recirculation pumps should incorporate a method of introducing water from thermal storage to the recirculation loop any time thermal storage temperatures exceed the DHW set point. This will reduce auxiliary heat requirements when DHW usage is low and excessive amounts of solar insolation are available. A further benefit is derived from the increased collector efficiency achieved as collector operating temperatures fall due to lower thermal storage temperatures.


Valve No. 1
Valve No. 2
S1
Collectors
Heat Exchanger
S2
Auxiliary
Boiler
Q_{IN}
S3
Load Heat
Exchanger
Pump No. 1
Pump No. 2
Pump No. 3
Pump No. 4
Double Wall
Heat Exchanger
Pump No. 5
Cold Water
Supply
Hot
Water
Delivery
Pre-heat
Tank
S4
Aux.
Hot
Water
Heater

Sensor Locations in a Typical Liquid-Heating System

FIGURE 4.2.1

If unsafe or excessive thermal storage temperatures can be reached during periods of high solar intensity and low heat utilization, a method of automatic heat disposal should be incorporated into the solar energy collection system. This mechanism should avoid excessive system pressures and temperatures and should minimize the occurrences of solar fluid blow-down.

4.2.3 Thermal Stratification

Temperature stratification in thermal storage should always be utilized to the advantage of the solar energy system. Water to be heated should be taken from the coolest sections of the thermal storage, that is the bottom. Heated water should be returned to the top of the thermal storage tank. Mixing in thermal storage should be minimized so that temperature gradients from the top to the bottom of the tank are maximized. Water for use should be taken from the top of the storage tank as this is the warmest segment. In a recirculating system, the water should be returned to the bottom of the thermal storage tank.

4.2.4 System Size

A general rule of thumb for a preliminary estimate of a solar energy collector system is that a space heating system should be sized to meet about 40% of the annual heating requirements, and water heating systems should be sized to meet about 60% of the annual water heating requirements. Final system sizing should be made on the basis of life cycle costing or other acceptable cost/benefit analysis methods.

In practice, there are often other factors affecting size decisions. These include the space available for mounting solar collectors, structural requirements, shading, orientation, load profiles and economic constraints. New construction can frequently be cheaply modified to accommodate optimum solar energy systems, while retrofit systems are restricted by existing conditions.

4.2.5 Collector Layout And Orientation

Although existing facilities may severely restrict the layout and orientation of the proposed solar energy system, these restrictions often are not as detrimental as might be expected. Variations of 10° to 15° from optimal slopes and azimuths have little impact on system performance. Depending on the energy usage profile, azimuths varying up to 60° from south may actually be beneficial to the system, by allowing reductions in thermal storage, or utilization of existing building features to reduce system costs, while not severely affecting solar contribution per dollar invested.

Collectors may be ground, wall or roof-mounted. Roof-mounting has the advantage of having fewer shading and vandalism problems. The designer should assess each application for the opportunity to incorporate solar collectors in an economical, maintenance free and aesthetically pleasing manner.

4.2.6 Roof Mounting

Placement of solar collectors on roofs may require a structural analysis of the roof. Support structures and associated plumbing will account for the majority of added weight. However, the major structural problem which may result from collector arrays on roofs is snow accumulation. Snow accumulation caused by solar collectors may be double the design snow loads in a given situation. For this reason, it is very important to perform a complete analysis to evaluate whether design loads will be exceeded.

Before a solar collector array is placed on a roof, all necessary roof repairs should be made. Re-roofing around collectors or a support structure, is considerably more costly and less effective than redoing an open roof.

Careful attention should be paid to tie down details. Collectors must be securely tied down to resist wind movement. Points of attachment and other roof penetrations must be waterproof and able to withstand movement due to wind, vibration and thermal expansion and contraction.

4.2.7 Pipe Sizing

Pipe sizing is a trade off between piping costs and pumping costs. A flow velocity of 1.5 m/s is the recommended maximum. Higher velocities may result in noisy pipes and excessive pumping energy requirements, as well as rapid wear. Some designer discretion is required in evaluating the cost/benefit ratios of various pipe sizes. High pumping requirements reduce the net energy gain by the system, which will increase the unit costs for the net energy gained. In well designed systems, electrical energy requirements of 5% of the solar heat delivered are possible while consumptions of 10% are common.

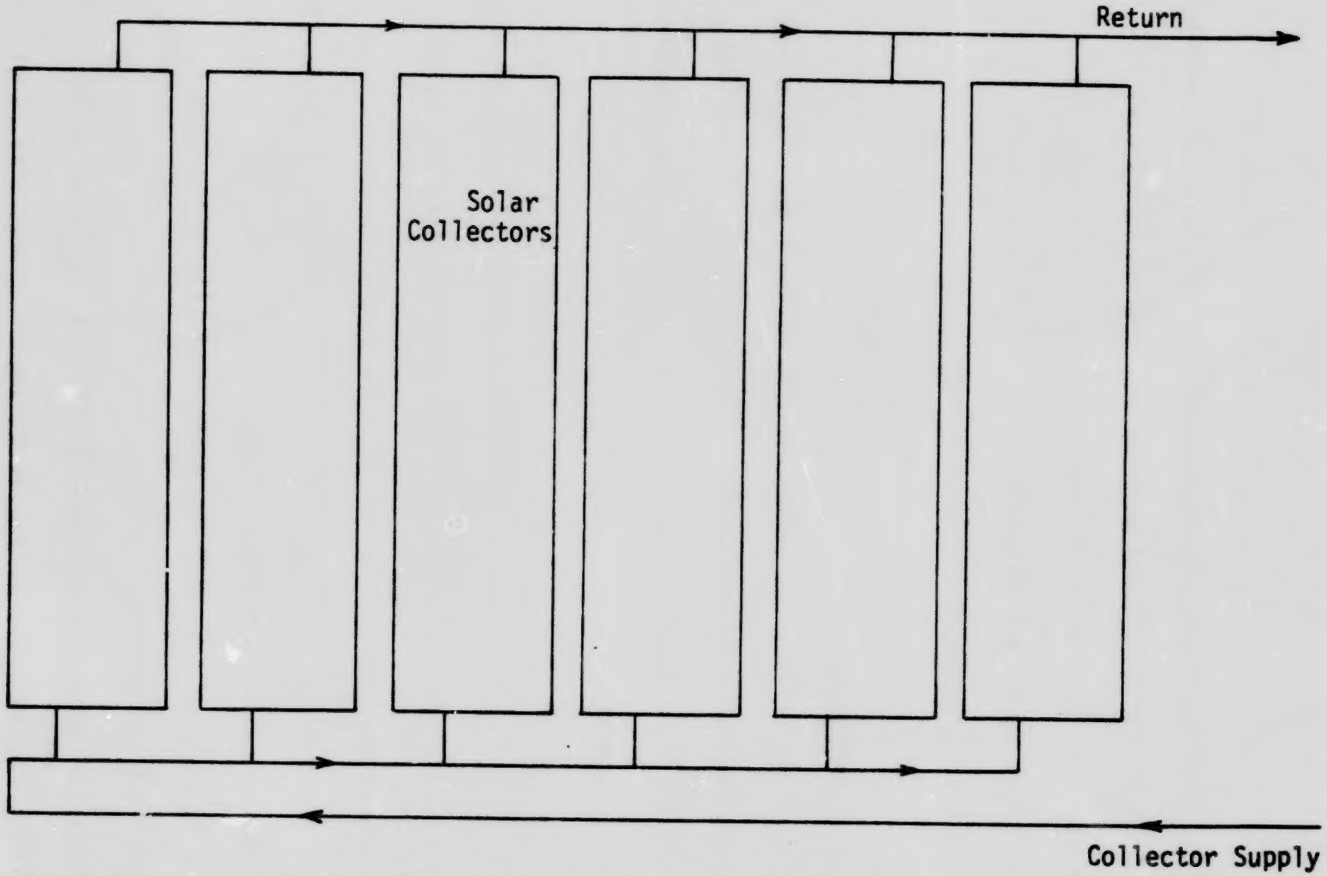
4.2.8 Piping Arrangements

Piping constitutes 10% to 40% of the costs in most solar energy systems, and therefore has the potential of significant cost savings. The length of pipe runs should be minimized to minimize cost, heat loss and piping requirements.

Piping should be graded to drain points to allow system evacuation for maintenance. If possible, the system should drain back to the tank which is used to fill the system. This will expedite the start-up after repairs are completed.

To induce balanced flow in all the collectors in an array, a piping arrangement called a reverse return is utilized. This arrangement (Figure 4.2.2) equalizes the pressure drop across the collectors and may eliminate the need for balancing valves.

Liquid collectors are generally plumbed for parallel flow. In some situations, it may prove desirable to use a series parallel arrangement in which a group of collectors is connected in series and then the groups in parallel. This will not affect the individual collectors' thermal performance provided the flow rate per unit area remains constant.


Return
Solar
Collectors
Collector Supply

Piping Arrangement To Equallize Collector Fluid Flow Rates Through All Solar Panels

FIGURE 4.2.2

For example, if three collectors, whose recommended flow rate is 5 litres per minute, are connected in series, then the flow rate should be increased to 15 litres per minute. Note however, that series flow will increase pump head losses.

Occasionally, a flow rate or collector fluid is desired which is different from that used to establish the collector's thermal efficiency curve. To be strictly accurate, the collectors performance parameters, $F_r\ (\ \alpha)_e$ and $F_r\ U_L$, should be adjusted. However, if the correction factors cannot be easily obtained, they can generally be neglected without serious error provided the actual flow rate is at least 80% of the test flow rate.

4.2.9 Thermal Expansion

The solar fluid and the components in direct contact with the solar fluid experience large temperature fluctuations on both a daily and a seasonal basis. Components must have the capability to absorb the expansion and contraction associated with these temperature fluctuations. Operating temperature fluctuations must be accommodated through the use of expansion compensators. Collector absorber plates must be allowed to expand in the collector box while the connections between collectors take up this expansion. Outdoor solar fluid piping may experience up to 0.5% expansion during a year and about 0.3% in a day. This is much more than domestic water systems undergo, and the requirements for expansion compensators is proportionally greater.

The thermal expansion of solar fluid is dependent on the fluid. Between its freezing point and boiling point, water undergoes a 4% volume change. Glycol solutions may have volume fluctuations of 9% per 100°C, which can amount to over 15% over the annual operating temperature range.

4.2.10 Galvanic Corrosion

An effort should be made to minimize the variety of metals in the solar energy system. Dielectric couplings should be used where dissimilar metals are joined. If severe corrosion problems are anticipated, as would occur in an aluminum-copper system, the installation of a sacrificial anode or an electrically induced galvanic protection system is recommended.

4.2.11 Plumbing Appurtences

Plumbing appurtences should be sufficient but not excessive. Strainers are necessary upstream of pumps. Valves should be located only where necessary for proper operation and maintenance procedures. By minimizing the number of valves, the possibility of inadvertently isolating a segment of the solar circuits is minimized.

Pressure gauges and thermometers located at key points in the system will permit easy system monitoring for trouble shooting and for performing system efficiency tests. Figure 4.2.3 shows the useful monitoring points in a liquid solar energy system. The use of Peterson plugs would permit a reduction in the cost and complexity of monitoring the system.

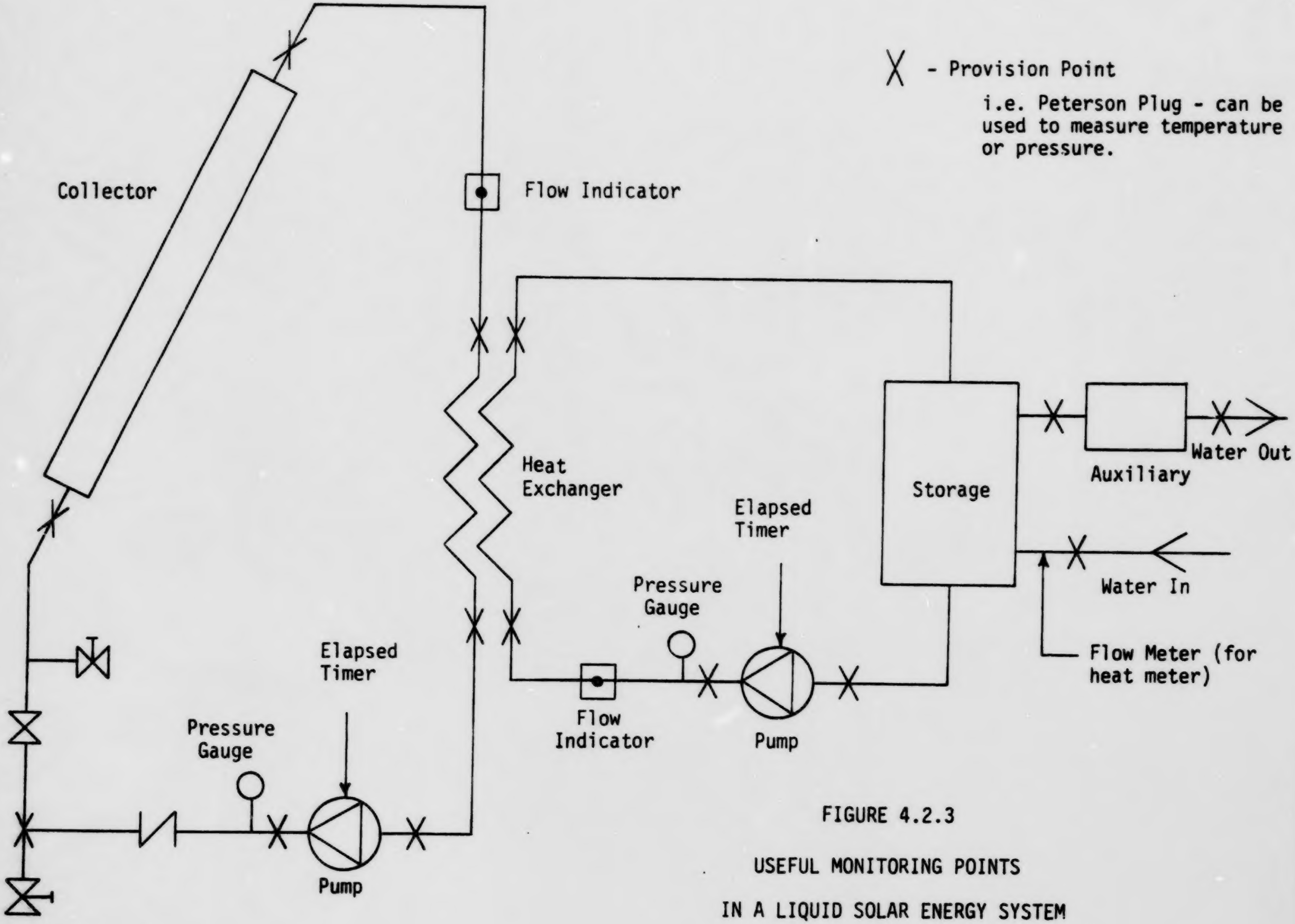

- Provision Point
i.e. Peterson Plug - can be used to measure temperature or pressure.
Collector
Flow Indicator
Heat Exchanger
Storage
Auxiliary
Water Out
Water In
Flow Meter (for heat meter)
Elapsed Timer
Pressure Gauge
Flow Indicator
Pump
Elapsed Timer
Pressure Gauge
Pump

FIGURE 4.2.3

USEFUL MONITORING POINTS

IN A LIQUID SOLAR ENERGY SYSTEM

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CHAPTER 5

AIR SYSTEMS

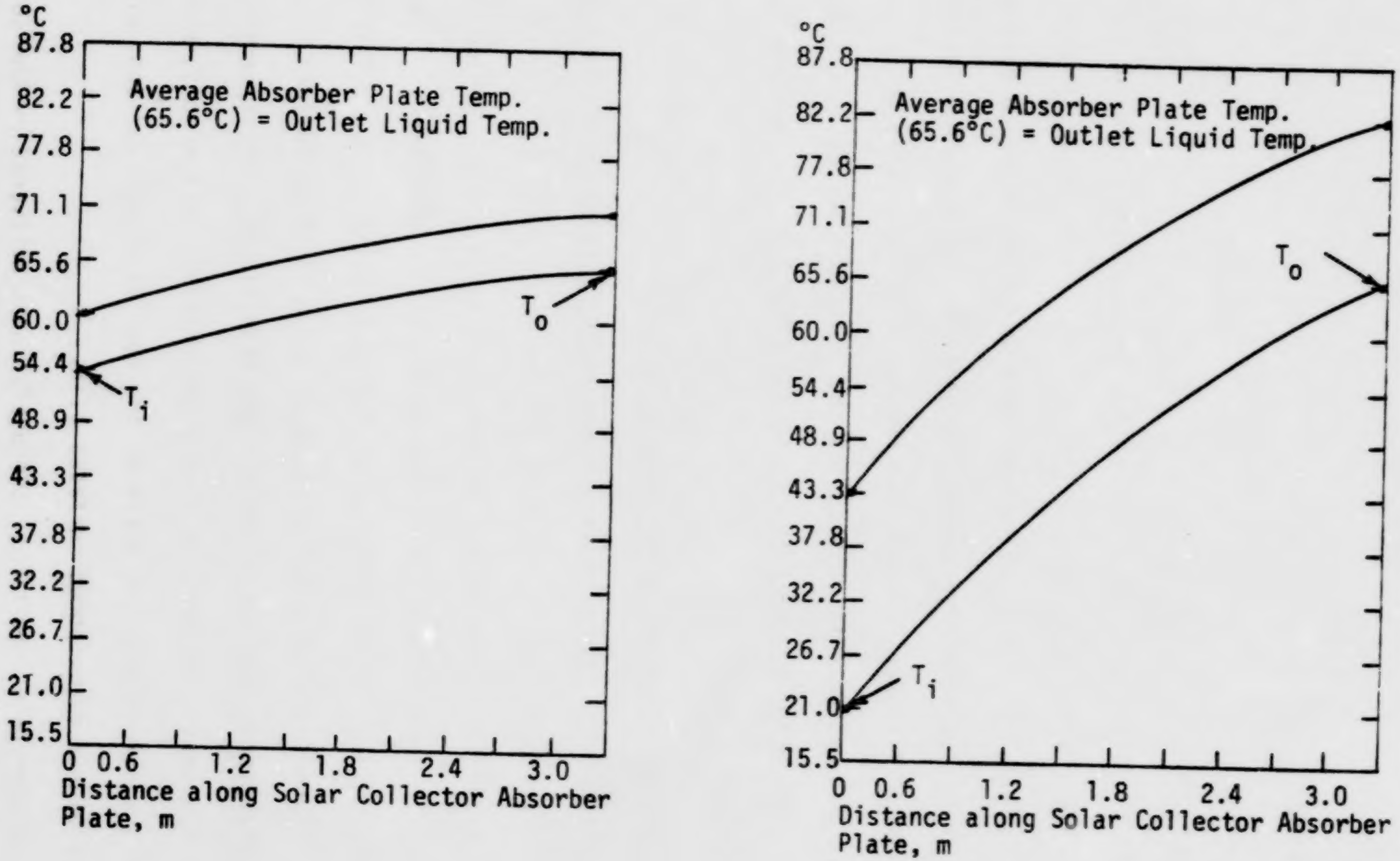
5.1 INTRODUCTION

Most of the flat plate solar collectors manufactured in Canada use a liquid as their working fluid. In the last few years though, production of air collectors has grown from zero to several hundred square metres per year. Solar energy system designers should therefore be familiar with the differences between the two types and should be conscious of what applications each is best suited to. This chapter discusses the performance and design of air collectors, and offers some guidelines for their proper application.

5.2 AIR COLLECTOR DESCRIPTION AND PERFORMANCE

The basis construction of most air and liquid collectors is the same, the major difference of course, being their choice of working fluids. A sketch of a typical air collector is shown in Figure 5.2.1. Notice that the air flows in direct contact with nearly the entire absorber surface and at a sufficient velocity for reasonably effective heat transfer. This is the major difference between the two types. Since fluid-to-surface heat transfer coefficients are much lower for air than liquids, it is necessary to increase the heat transfer area in air collectors. As a result, most designs use the entire absorber plate as the heat exchange surface, often with the assistance of extended surfaces (fins).

Under steady-state conditions, the useful heat delivered by the solar collector is equal to the radiant energy converted into heat at the absorber surface minus the heat losses from that surface. Expressed mathematically,

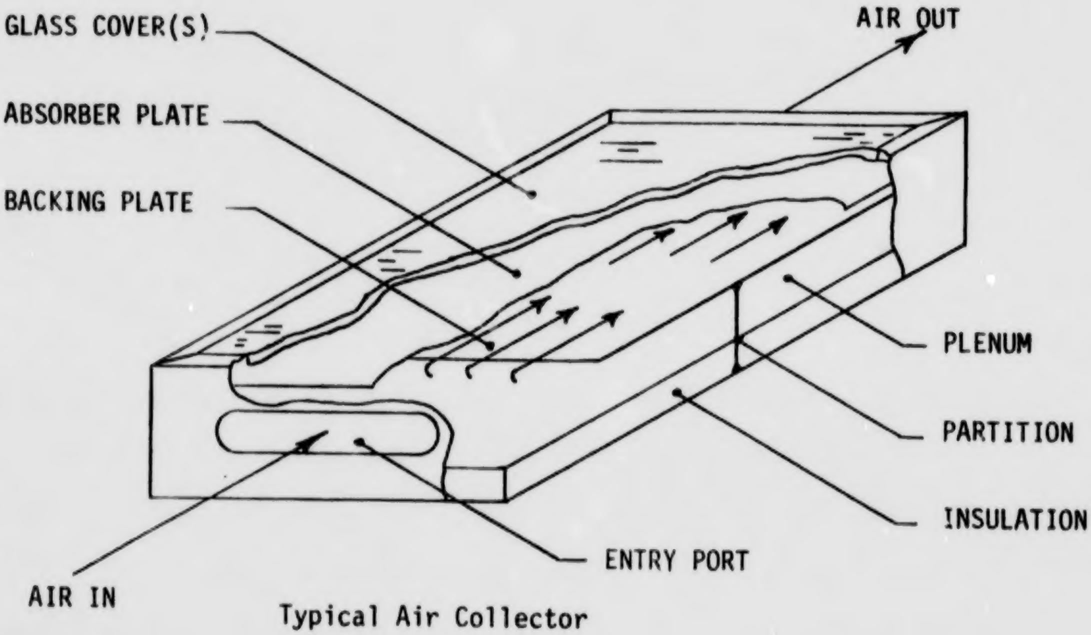


Air Heating Solar Collector

Air Heating Solar Collector

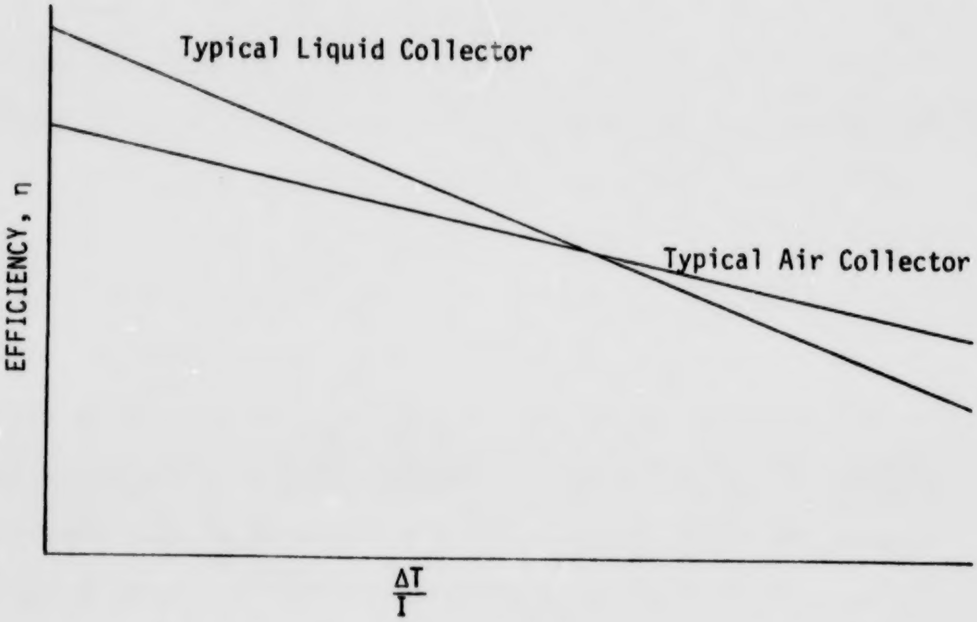
Comparison of Typical Temperatures in Liquid and Air Heating Solar Collectors

FIGURE 5.2.2


GLASS COVER(S)
ABSORBER PLATE
BACKING PLATE
AIR OUT
PLENUM
PARTITION
INSULATION
ENTRY PORT
AIR IN

Typical Air Collector

FIGURE 5.2.1



Comparison Of Typical Thermal Efficiencies
Of Air And Liquid Collectors

FIGURE 5.2.3

this is the familiar equation

$$Q_u = A_c F_r [(\tau\alpha)_e I - U_L (T_i - T_a)]$$

where Q_u = useful heat gain

and $F_r = \frac{\text{heat removed by collector}}{\text{heat removed if all fluid were at } T_i}$

= collector heat removal factor

Due to the low specific heat of air (approximately one-fourth that of water), F_r for most air collectors will range from 0.6 to 0.7. In contrast, F_r for liquid collectors is typically 0.9 or above. Since removal of equivalent amounts of heat from a collector (at equal mass flow rates) will cause air to experience a temperature rise four times that of water, the average operating temperature of an air collector will be higher than a liquid collector's at equal inlet temperatures and mass flow rates. This is shown in Figure 5.2.2. As a result, heat losses are greater for the air collector and F_r is lower. The smaller value of F_r means that the Y-intercept of the standard thermal efficiency curve, $F_r(\tau\alpha)_e$, will be lower, while the slope, $F_r U_L$, will be shallower than a typical liquid collector's is. This is shown on Figure 5.2.3.

This does not necessarily imply that liquid collectors produce greater amounts of energy than air collectors, but rather that the collector must be carefully matched to the anticipated load. For example, at high values of $\Delta T/I$, an air collector may offer superior energy production. Another factor to consider is that heated air can be delivered at much lower temperatures and still be of value for space heating purposes. Thus, an air system may be able to operate at a lower $\Delta T/I$ than a liquid system and still do the same job.

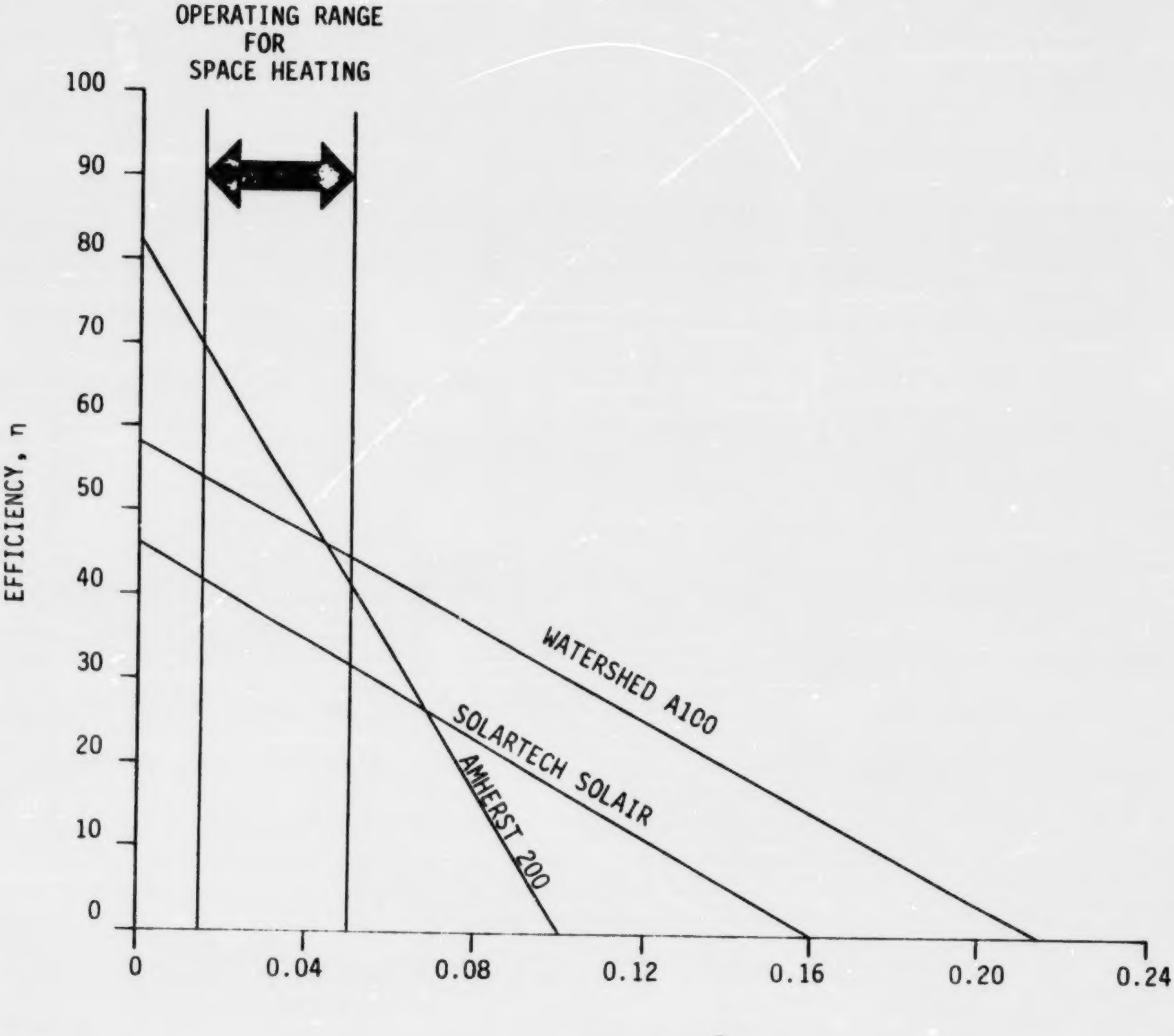

OPERATING RANGE
FOR
SPACE HEATING
EFFICIENCY, η
100
90
80
70
60
50
40
30
20
10
0
0
0.04
0.08
0.12
0.16
0.20
0.24
WATERSHED A100
SOLARTECH SOLAIR
AMHERST 200
$\frac{T_i - T_a}{I} \frac{C - m^2}{W}$

FIGURE 5.2.4

A comparison of efficiency curves for a number of air collectors is shown in Figure 5.2.4. Notice the wide variation between different designs. The Watershed collector, no longer in production, is shown as a dashed line because of the excessive scatter in the data from its efficiency test. The prototype overlapped plate was an experimental design intended to reduce the pressure drop through the collector; it has yet to enter production. The Solaron is an American collector which is produced in large quantities and probably represents the typical medium performance collector with acceptable reliability and durability. Both the Solartech and Amherst collectors are Canadian-made and currently in production. The Suntrap, also a Canadian design, is no longer in production.

Stagnation temperatures of most air collectors are approximately 100°C to 140°C. Their typical weight is 27 to 90 kg/collector or 9 to 40 kg/m^2.

Obviously, the final selection of a collector must consider other factors than just thermal performance. Cost, life, durability, availability, etc. all have to be investigated before the final selection can be made.

5.3 APPLICATIONS

The three most common applications of air systems are space heating, ventilation air preheating and crop drying.

5.3.1 Space Heating

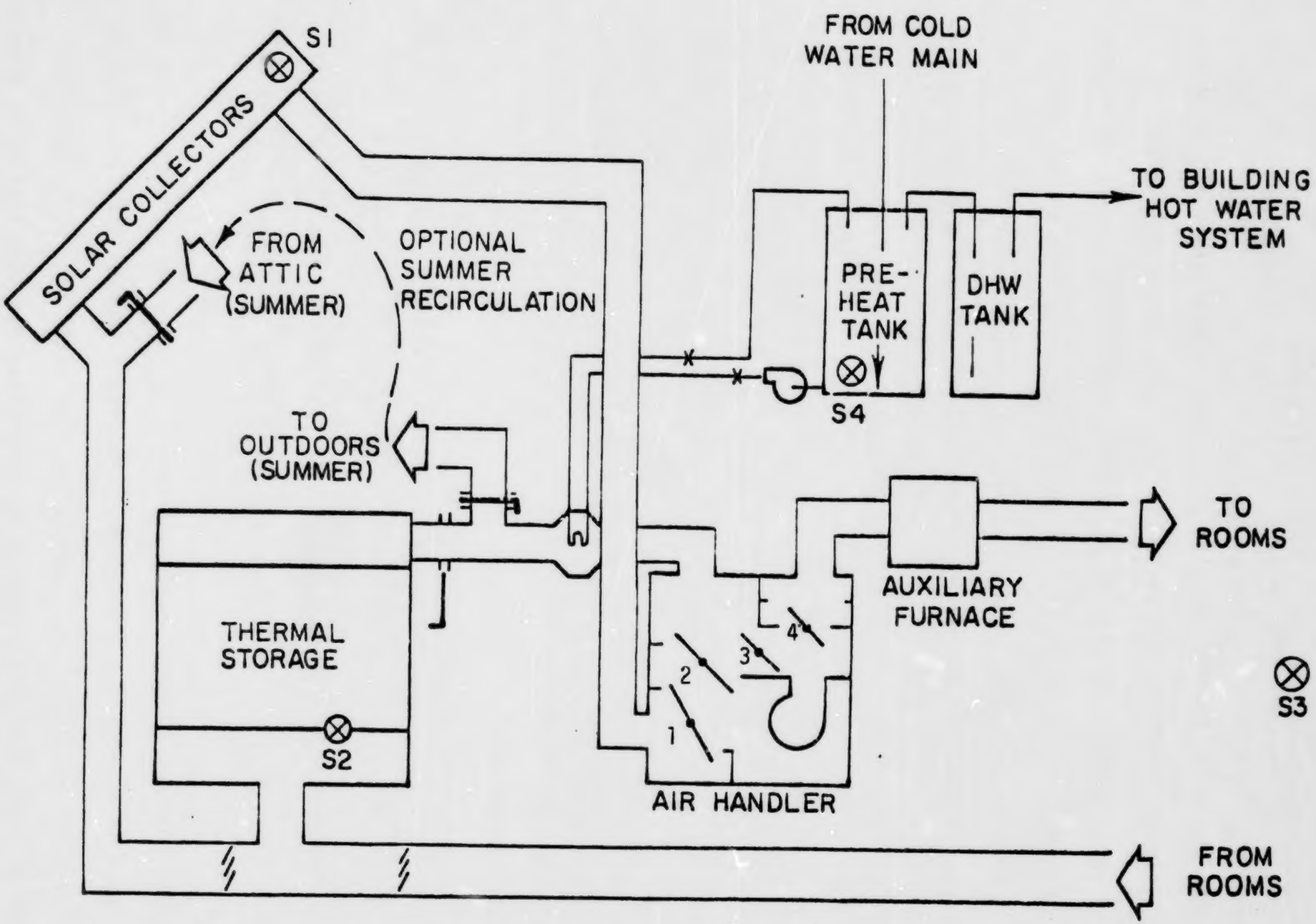
This has been the traditional use of most air collectors. Space heating applications generally require a full complement of auxiliary components, including storage, complex controls, and a separate air handling system. Inlet temperatures are typically 10°to 40°C above ambient and the operating efficiency ranges around 30% to 70%. Due to the comparative sophistication of the systems and their less than optimum operating regime (discussed in more detail below), space heating with air collectors can, at best, be described as only a fair application of solar energy in the Canadian environment.

A typical airspace heating system is shown in Figure 5.3.1. Its major components are the: collectors, pebble bed storage (shown in more detail in Figure 5.3.2), air handler, DHW preheat tank with coil and associated ducting. Note the position of the temperature sensors. The design shown uses a single blower and is intended primarily for residential applications. It has three major modes of operation which are controlled by the position of the motorized dampers in the air handler:

Mode of Operation	Damper Position			
	A	B	C	D
Collectors delivering heat to storage	open	closed	open	closed
Collectors delivering heat to rooms	open	closed	closed	open
Storage delivering heat to rooms	closed	open	closed	open

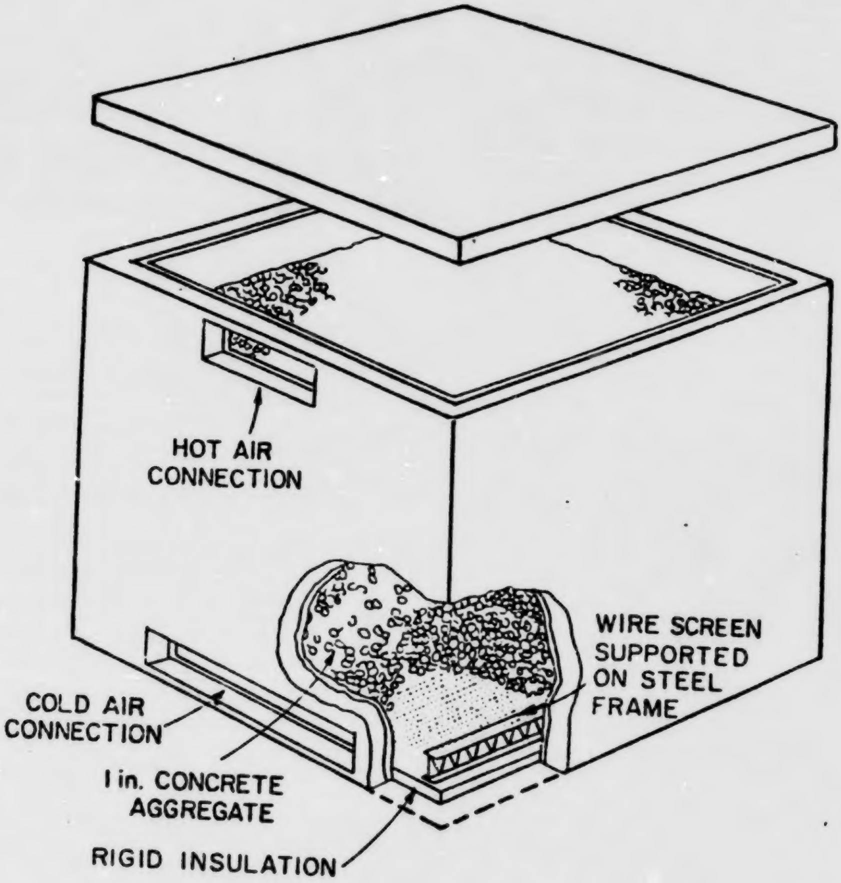
An optional summer recirculation mode for preheating the domestic hot water (and cooling the collectors) is also available using the three manual dampers shown.

Notice that the storage is always charged in one direction and discharged in the opposite direction. This allows significant thermal stratification to be developed in the pebble bed. Due to the poor heat transfer between adjacent pebbles, this stratification can be maintained for long periods of time.


S1
SOLAR COLLECTORS
FROM ATTIC (SUMMER)
OPTIONAL SUMMER RECIRCULATION
TO OUTDOORS (SUMMER)
THERMAL STORAGE
S2
FROM COLD WATER MAIN
PRE-HEAT TANK
S4
DHW TANK
TO BUILDING HOT WATER SYSTEM
1
2
3
4
AIR HANDLER
AUXILIARY FURNACE
TO ROOMS
S3
FROM ROOMS

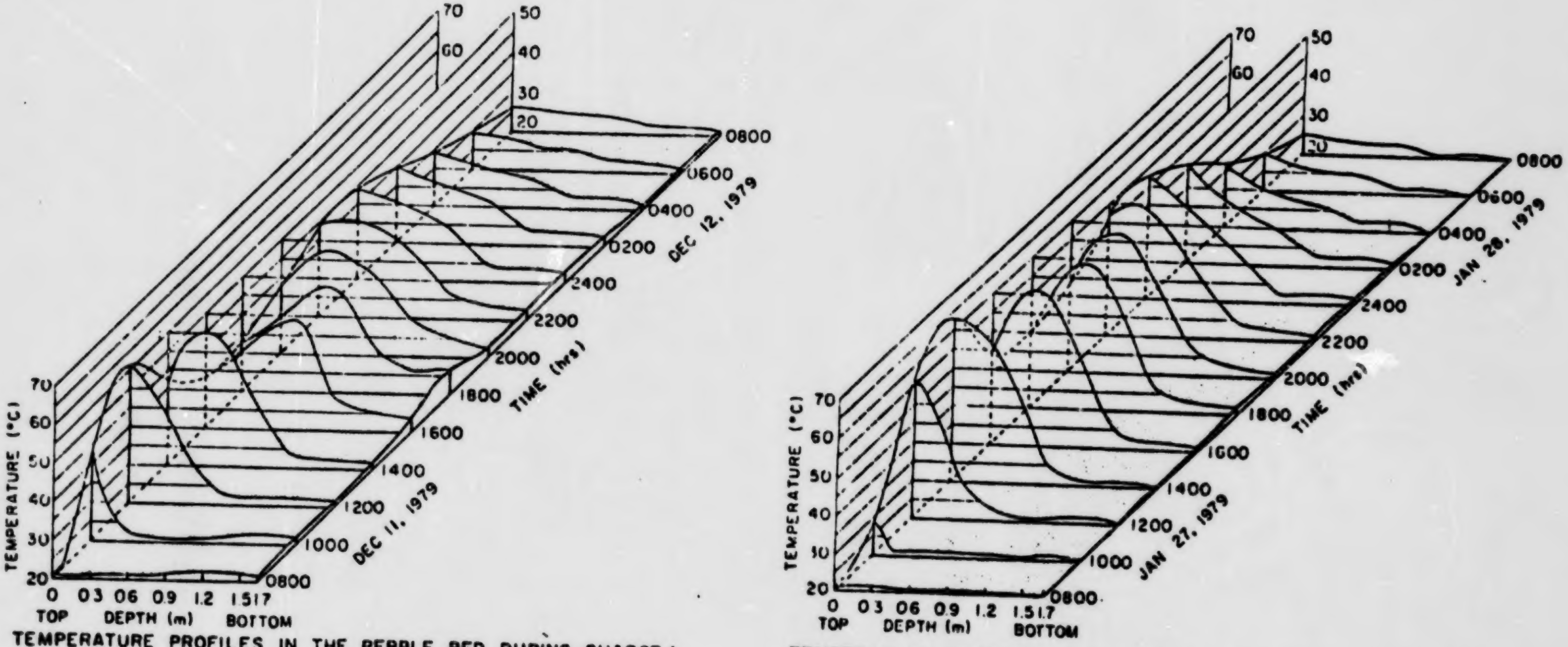
TYPICAL AIR SPACE HEATING SYSTEM

FIGURE 5.3.1


HOT AIR CONNECTION
WIRE SCREEN SUPPORTED ON STEEL FRAME
COLD AIR CONNECTION
I in. CONCRETE AGGREGATE
RIGID INSULATION

PEBBLE-BED HEAT STORAGE UNIT

FIGURE 5.3.2



TEMPERATURE PROFILES IN THE PEBBLE BED DURING CHARGE/ DISCHARGE CYCLE AIR FLOW RATE 13.1 ℓ/(sec m²)

TEMPERATURE PROFILES IN THE PEBBLE BED DURING CHARGE/ DISCHARGE CYCLE AIR FLOW RATE 9.6 ℓ/(sec·m²)

TYPICAL TEMPERATURE STRATIFICATION IN A PEBBLE BED

FIGURE 5.3.3

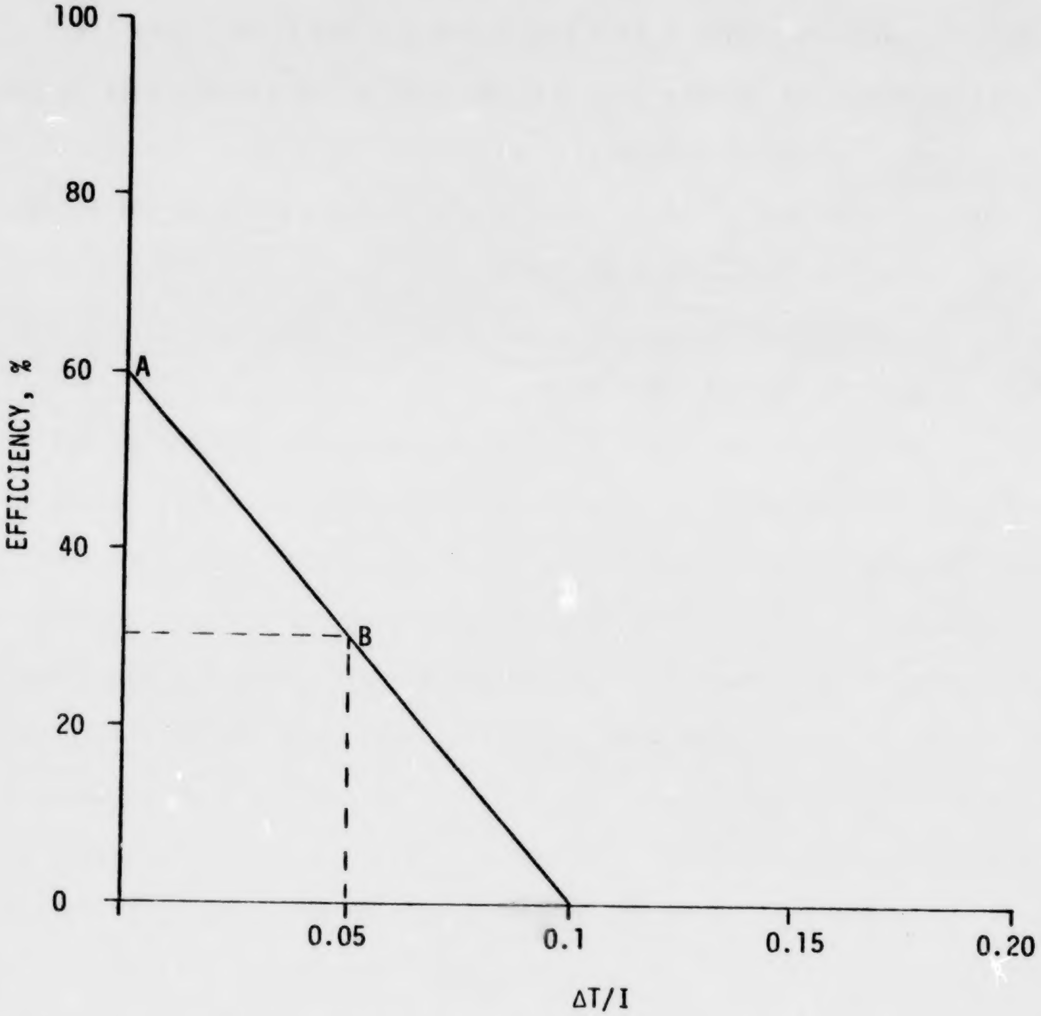
Typical temperature profiles in a pebble bed are shown in Figure 5.3.3.

It is important to note that pebbles used in the storage must be thoroughly washed to remove all dust.

Other, nonresidential, space heating applications which do not use any thermal storage have also been designed.

5.3.2 Ventilation Air Preheating

Many buildings, especially large commercial, industrial and agricultural structures, have significant ventilation air heating requirements. Because of their need to exhaust contaminated or moisture-laden air, fresh air has to be introduced into the structure and heated. Ventilation air preheating is a promising application for air collectors since they can operate at a high (and constant) efficiency, equal to the Y-intercept on the efficiency curve (since $\Delta T/I = 0$). Figure 5.3.4 demonstrates the thermal advantage of such an arrangement. System costs are generally quite low since no storage, automatic control system or mounting structure (if a suitable south-facing wall is available) is required. Since many applications are factory environments, trained maintenance personnel are normally available to service the system. Note that this may not necessarily apply to structures which have significant quantities of internal heat generation with comparatively small fresh air requirements, such as office buildings. In these cases, outdoor air is used not only for ventilation purposes, but also to prevent overheating of the building's core. In some applications with moderate internal heat generation, such as schools, a control system may be required which balances the (solar) preheated air with the incoming outdoor



The thermal efficiency curve shown is typical of many air collectors. Notice that by operating at point A (preheating), the efficiency is twice that available at point B (space heating), 60% versus 30%.

Importance of Proper Application For Efficient Collector Operation

FIGURE 5.3.4

air to provide the desired mixed air temperature. Since ventilation rates on many large commercial and industrial structures are often quite high, the designer should ensure that the collector's maximum flow rate is not exceeded.

The most desirable applications are those which have their entire requirement for outdoor air occurring during daylight hours, such as in a single shift factory. Then, the demand for preheating energy will occur simultaneously with the periods of available solar radiation. Ventilation air preheating is generally more economic on new construction than retrofits.

Of course, heat recovery may also be considered as a means to reduce the preheating load. Since heat recovery systems are easiest to install when the exhaust and outdoor air streams are adjacent to each other, solar preheating can best compete against heat recovery when the outdoor air inlets and exhaust air outlets are not physically in close proximity to each other.

On some installations, it may be necessary to filter the air prior to admitting it to the collectors.

5.3.3 Crop Drying

Many agricultural crops, such as grainery rice and tobacco, are harvested in a semi-moist condition. Drying is completed using low temperature, high volume air flows with the heating energy often being supplied by expensive fuels such as propane. Solar energy can be used to supply a portion or all of this load with the application being rather similar to ventilation air preheating. The major advantage of using air collectors, of course, is the high efficiency at which even simple, inexpensive models can operate.

The major disadvantages is that the drying system can only be used for a small fraction of the year, typically one to three months. Unfortunately the drying period also occurs during the fall and early winter months when available solar radiation is at a minimum.

5.4 DESIGN PROBLEMS

The design of any solar collector, whether air or liquid, requires careful consideration of many, often conflicting, requirements such as cost, durability, performance, weight, etc. These common design criteria are discussed in more detail in Chapter 6. However some of the unique problems which air collectors experience are described below.

1. Because of the low absorber-to-air heat transfer coefficients, it is necessary to significantly increase the heat transfer surface area and thus enlarge the total wetted area of the flow passage. This makes proper sealing of the flow channel(s) more difficult, with the result that air leakage into or out of the collector may seriously degrade performance. Measurements of air collectors have indicated that leakage may equal 5% to 10% (or more) of the total collector air flow rate. Proper sealing

of the flow passages is often difficult and expensive. As a result, a thorough check of the collectors (and the system), to find any leaks, should be carried out prior to final commissioning.

2. Degradation of the critical absorber plate surface due to condensation or abrasion may occur if the airflow is passing over the front of the plate.

3. Corrosion of the air flow channel(s) due to condensation may also result when the system is shut down and the collector is allowed to cool.

4. The use of air collectors generally necessitates the use of a good air filtration system, especially if the pebble bed storage system is used.

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CHAPTER 6

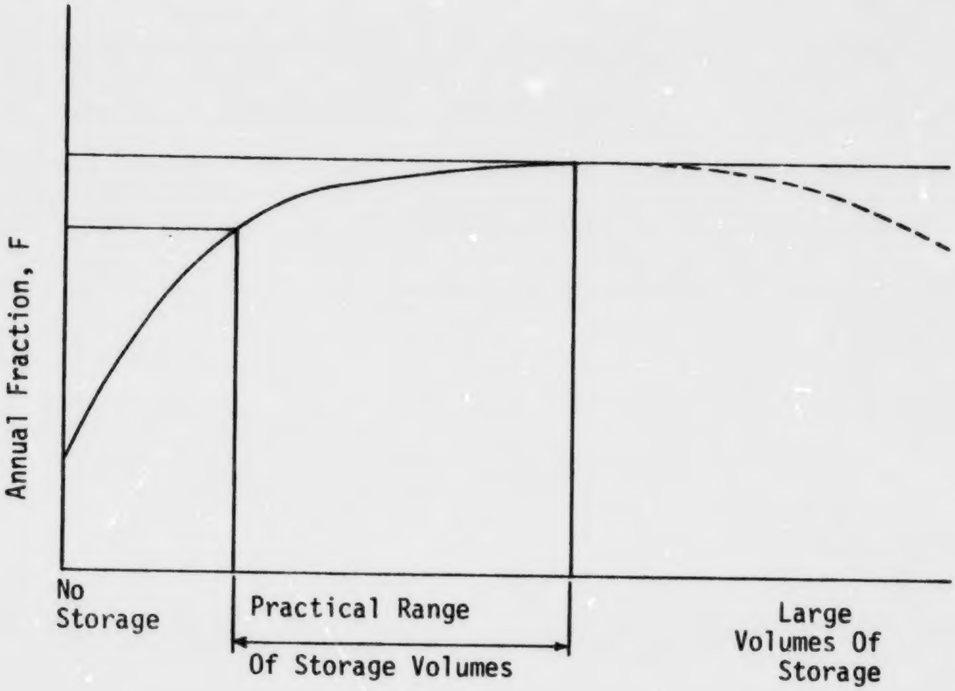
SYSTEM COMPONENTS

6.1 HEAT STORAGE

The purpose of heat storage is to capture and store excess solar energy when it is available and make it available for use when the incident solar energy cannot meet the heat load. The heat may be stored by raising the temperature of the storage medium (sensible heat), changing the phase of the storage medium (latent heat) or by utilizing certain chemical reactions. The heat is recovered by reversing the storage operation. Long term thermal storage, although of interest, is still experimental and is not covered in depth here.

6.1.1 Sizing Thermal Storage

The effect of storage size on solar heat contribution is shown in Figure 6.1.1. With no storage, only energy collected during times of energy demand can be utilized. Storage permits a delay between the time of collection and utilization of the solar energy. At first the increase in solar fraction rises rapidly as storage size increases, but it increases at a diminishing rate. As storage is increased beyond a certain size in a DHW system, solar fraction begins to decrease. This is due to a decrease in storage temperature which results in a lower availability of useful energy. Also skin losses from the storage tank increase with surface area. Thus, excessive storage volumes actually degrade the performance of DHW systems.



Effect of Storage Size on Solar Heat Contribution to a DHW Load

FIGURE 6.1.1

The exact amount of storage required is dependent on the heating load profiles. In a residential hot water system where most of the load occurs in the morning and the early evening, when incident radiation is usually low, storage is a critical component in the solar energy system, and should be sized to hold about one day's hot water requirements.

A processing plant which has a continuous hot water draw and a modestly sized solar energy system would not require any solar storage because the energy would be utilized as it was collected.

For most applications, the practical range of thermal storage is 200 to 300 kJ/°C m^2 of collector area (50 to 75 litres of water storage per square meter of collector). A rule of thumb is to size storage to meet a one day heat load or one day's incident radiation.

Water is the most common storage medium used in liquid systems because of its price, good heat storage capabilities, and its consistency of viscocity and thermal properties in the relevant range.

The solar fluids used in the solar collector loop can be used for thermal storage, but these are usually expensive and have less favourable thermal, chemical and mechanical properties than water. Tables 6.1.1 and 6.1.2 give properties and comparisons of some typical solar fluids while Table 6.1.3 lists some by trade name.

Fluid storage tanks can be made of a variety of materials and in several configurations. The design will depend on the location of the storage, and the storage temperature and pressure. In all cases, storage should be designed to promote thermal stratification, which can be bene-

Table 6.1

EXAMPLES OF TYPICAL HEAT TRANSFER LIQUID PROPERTIES

Property	Water	Glycols		Silicone Fluid	Hydrocarbons		Glycerine ("Glycerol") 60%/40% Glycerol/water
		50% Ethylene Glycol/Water	50% Propylene Glycol/Water		Aromatics	Paraffinic Oil	
Freezing point, (°C)	0	-36	-33	-50	-62 to -9	-9	-35
Boiling point, (°C)(at atm. pressure)	100	110 (197 for 100% glycol)	106 (180 for 100% glycol)	None	181-337	371	110
Fluid Stability	Requires pH or inhibitor	Requires pH or inhibitor monitoring	Requires pH or inhibitor monitoring	Good	Good to Fair	Good to Fair	Good
Flash Point (°C)	None	None (111 for 100% glycol)	None (99 for 100% glycol)	232-315	56-207	149-235	None (177 for 99% glycerol)
Specific Heat 23°C (KJ/KG/°C)	4.18	3.35	3.56	1.42-2.01	1.51-2.09	1.80-2.64	3.35
Thermal Conductivity at 38°C kW/cm °C	6.28×10^{-3} @ 0°C	3.98×10^{-3}	3.89×10^{-3}	1.44×10^{-3}	1.21×10^{-3}	1.21×10^{-3}	4.67×10^{-3}
Kinematic Viscosity $\times10^{5}$ m^2/s at 25°C	0.083	0.316	0.464	1.86-4.64	0.744-4.64	0.093-5.58	0.651
Toxicity	Depends on inhibitor used	Moderate to high depends on inhibitor used	Slight depends on inhibitor used	No unequivocal toxic effects are recognized	Moderate	Low	Slight
Lifetime Estimated	Depends on inhibitor used	Less than 2 years	Less than 2 years	More than 5 years	Ranges 1-20 years	On order of 10 years	More than 5 years

TABLE 6.1.2

GENERAL COMPARISON OF TYPICAL HEAT EXCHANGER FLUIDS

FLUID	ADVANTAGES	DISADVANTAGES
Water, untreated	1. nontoxic 2. environmentally safe 3. inexpensive 4. high thermal efficiency	1. freezes at 32°F 2. supports galvanic corrosion 3. boils at 212°F 4. promotes scale formation 5. may change composition in closed system
Water/glycol mixtures	1. will not freeze down to -35°F 2. inhibitor technology for ethylene glycol well-suited to offering protection in a multi-metal system 3. boils at temperatures higher than water	1. fluid must be replaced frequently or maintained properly (by monitoring ph) 2. problem of disposal of large amounts 3. decomposes around 280 - 300°F forming sludge and organic acids 4. toxic (ethylene glycol) 5. can degrade building material (roofs)
Hydrocarbons	1. low cost 2. nonvolatile 3. less toxic than ethylene glycol	1. poor oxidation stability 2. poor thermal stability at high temperatures resulting in sludge and acid formation 3. high viscosity at low temperatures 4. environmental effects similar to motor oil 5. typical closed-cup flashpoints run 300 - 320°F, fluids with higher flashpoints have a higher viscosity 6. damage to building from leakage may be irreparable due to odor and low volatility 7. low heat conductivity
Silicones	1. do not freeze 2. do not boil 3. do not corrode common metals 4. long life 5. high flashpoint 6. low toxicity, if any	1. low heat capacity 2. high viscosity 3. expensive 4. not biodegradable 5. difficult to seal 6. strong tendency to leak 7. may require more expensive pump (may also be true for hydrocarbons)

Based on mfr. claims

TABLE 6.1.3

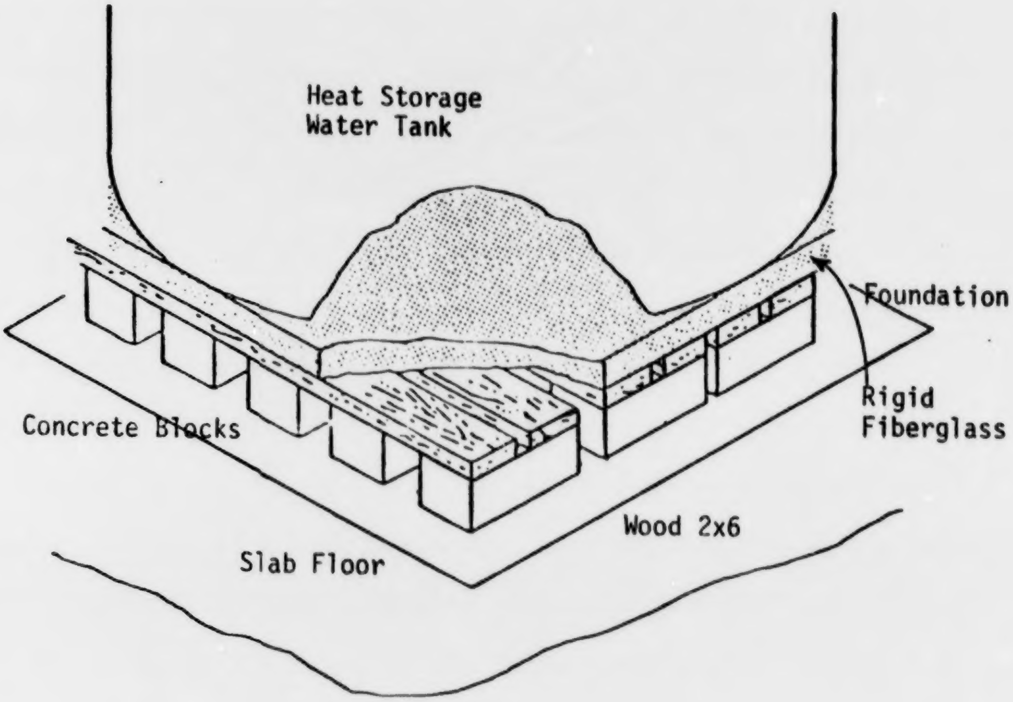
SOLAR HEAT TRANSFER FLUIDS

TRADE NAME	MANUFACTURER	COMPOSITION
•UCAR-17	Union Carbide	Ethylene glycol
•DOWTHERM SR-1	Dow Chemical	Ethylene glycol inhibitor < 5% and red dye < 1%
UCON-500	Union Carbide	Polyglycols - may contain phenyl β naphthyl amine < 10%
•UCON-30	Union Carbide	Ethylene oxide
•DOWFROST	Dow Chemical	Propylene glycol - contains inhibitor
•UCAR-35	Union Carbide	Propylene glycol
MOBILTHERM 603	Mobil Oil	Paraffinic, neutral oil
MOBILTHERM LIGHT	Mobil Oil	Paraffinic, high aromatic oil
SUNTEMP 1	Resource Technology Corp.	Paraffinic, low aromatic oil contains red dye (< 1%)
CALORIA HT-43	Exxon	Paraffinic, low (14%) aromatic oil
THERMIA 33	Shell	Paraffinic oil - aromatic content 20 - 25%
PROCESS OIL-3029	Exxon	Naphthenic mineral oil aromatic content < 10%
•DOW CORNING 200	Dow Corning	Silicone oil
SF 96	General Electric	Polydimethylsiloxan
•THERMINOL 66	Monsanto	Terphenyl mixture
•DOWTHERM A	Dow Corning	Diphenyl and diphenyl oxide mixture
SOLARGARD G	Daystar	Glycerol (USP) 60%/water 40%

ficial to system efficiency.

The most desirable location for a storage tank is indoors. Outside and underground storage tanks are subject to corrosion and deterioration of insulation while heat losses from indoor tanks can meet part of a building's heat requirements. For high temperature or high pressure storage, steel tanks are recommended. Commercial grade hot water pressure storage tanks are available in a variety of configurations and sizes but are quite expensive. Light gauge metal tanks, fiberglass reinforced plastics (F.R.P.) tanks and wood or masonary tanks lined with plastic membranes are moderately priced alternatives for unpressurized thermal storage. It may be convenient to utilize foundation walls for one or more of the tank sides. Tanks that are constructed in place have the advantage that their dimensions are not restricted by the size of doorways and passages. Square or rectangular storage tanks can be easily and neatly insulated. Storage tanks should be insulated from direct contact with walls and floors. Tank linings or waterproof membranes should be flexible and strong enough to resist damage if the tank walls or footings shift or crack. The tank and its membrane should be accessible for repairs or modifications, should the need arise.

Insulation levels of RSI5 (R30) are recommended for outdoor thermal storage and RSI2 (R 10-12) is recommended for indoor thermal storage. Flat bottomed tanks should sit on rigid insulation to reduce bottom losses. (Figure 6.1.2) Saddle supports under cylindrical tanks are quite difficult to insulate, but the effort is worthwhile. Care should be taken to


Heat Storage
Water Tank
Foundation
Rigid
Fiberglass
Concrete Blocks
Wood 2x6
Slab Floor

Bottom Insulation and Support Scheme for Flat-Bottom Water Storage Tanks

FIGURE 6.1.2

ensure that insulation on the floor does not stand in water, as wet insulation may lose insulating properties.

Underground tanks must be insulated with a material that does not absorb moisture and does not degrade. Tanks should be located below the frost line and above the water table, which precludes their use in most Canadian localities. If possible, underground installations should be avoided.

A method to drain or pump out tanks should be included in the storage system design. Dielectric connections should be made where piping and tanks are of dissimilar metals.

Provision must be made for expansion and contraction of the liquid storage. In open storage tanks this involves leaving sufficient freeboard to accommodate any thermal expansion, or providing an overflow drain. In closed pressure systems, expansion tanks or pressure relief valves are necessary to avoid damage caused by excessive pressures. (Figure 6.1.3)

Provisions must also be made to permit disposal of excess heat. In open systems, permitting the water to boil is one solution. This results in a buildup of scale in the storage tank and necessitates having a make-up water system. The high temperature can also be hard on the collectors.

It is important in pressurized systems to incorporate temperature relief valves in the storage system. If the "blow off" temperature is much above 100°C, the water will flash to steam as it leaves the tank. This is a hazard requiring special consideration. By blowing off hot storage water and introducing cold make-up water, safe storage temperatures

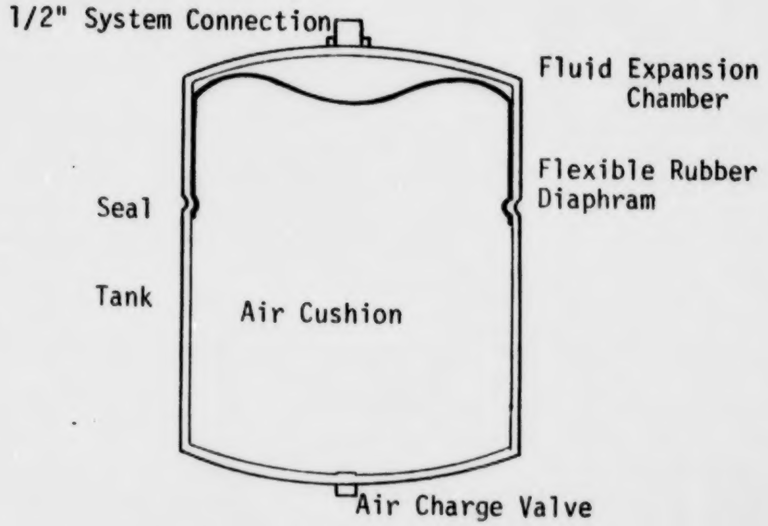

1/2" System Connection
Fluid Expansion Chamber
Flexible Rubber Diaphram
Seal
Tank
Air Cushion
Air Charge Valve

FIGURE 6.1.3

can be maintained.

Other methods that can be used to avoid excessive storage temperatures include:

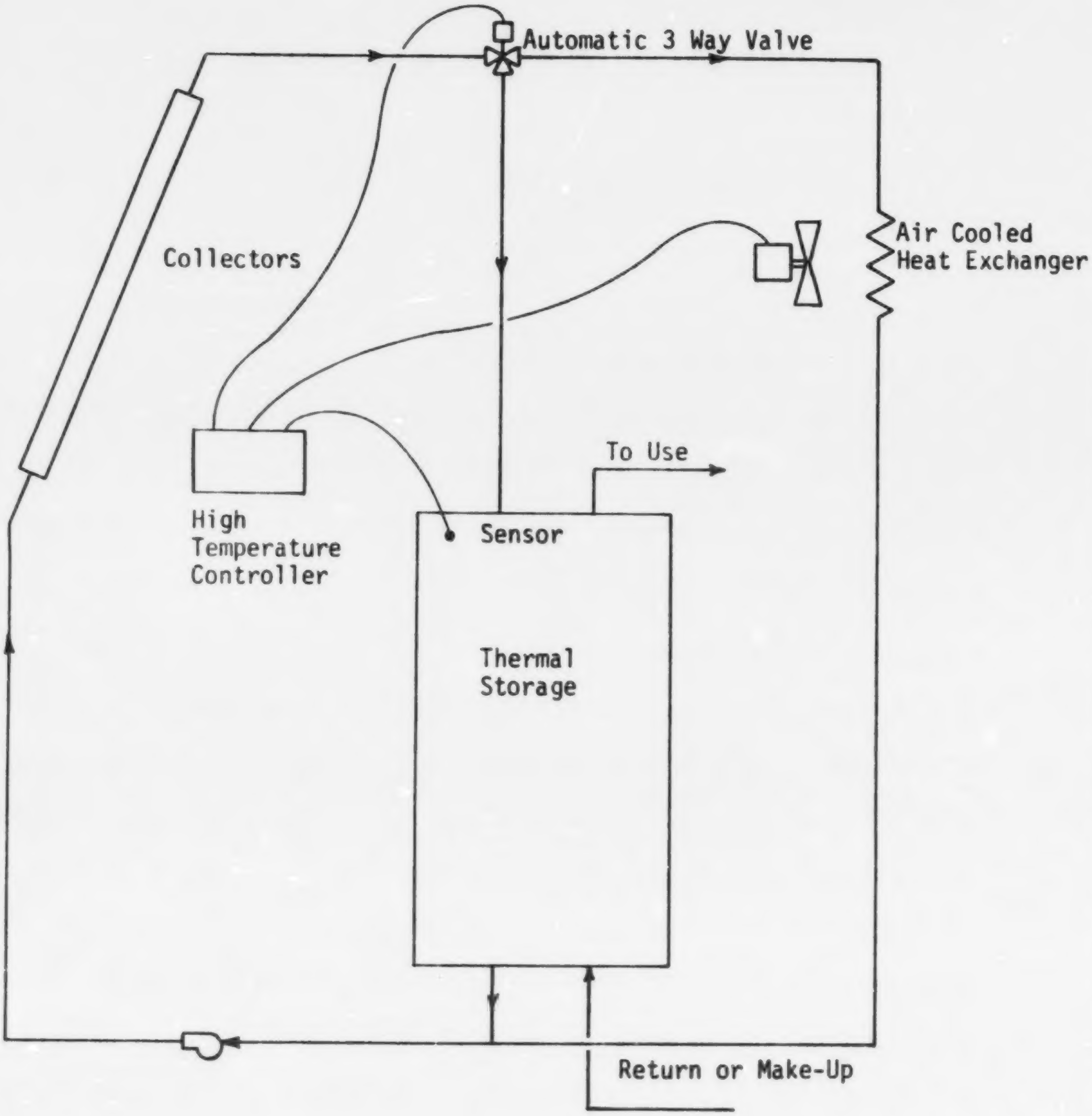
(1) having a fan coil arrangement which cools the fluid with outside air (Figure 7.1.4).

(2) having a high temperature shut-off on the collector pump, which would result in boiling in the collectors instead of in storage. The collectors must be able to stand stagnation temperatures and be equipped to allow the steam to escape.

When boiling in the collector or in storage is expected, water treatment with softeners and corrosion inhibitors is recommended. Glycol solutions are not recommended for use above 150°C for at these temperatures, chemical breakdown into acids is accelerated.

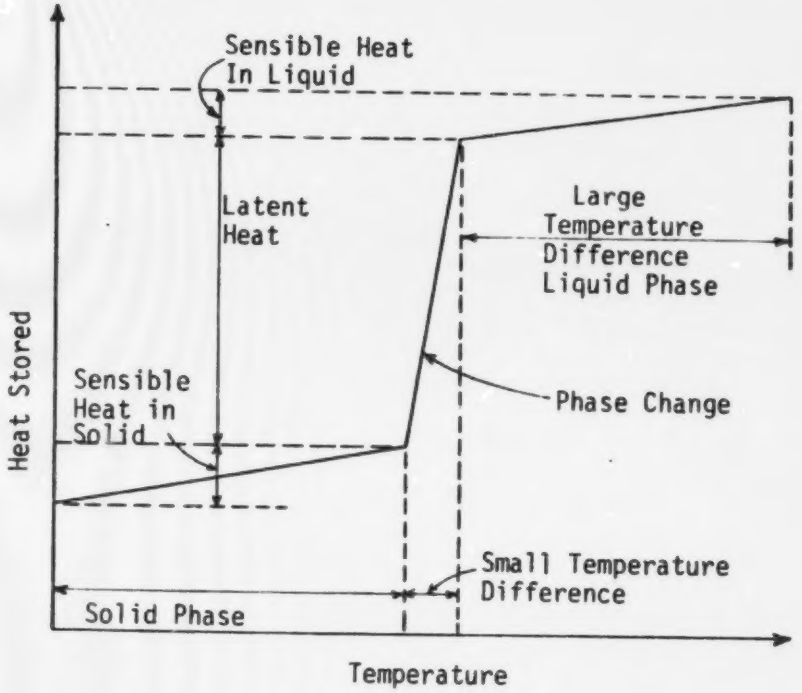
6.2 PHASE CHANGE MATERIALS (PCM's)

As their name suggests, PCM's utilize the latent heat of fusion for thermal energy storage with the heat being stored by liquifying the material (Figure 6.2.1). The advantages of PCM's are the reduced storage volume required to store a given quantity of heat, and the near constant temperature of the solar storage. Relatively low storage temperatures are conducive to high collector efficiency and constant temperatures simplify system design. In passive systems, low temperature PCM's are used to prevent the building space from overheating during the day and from excessive cooling at night. In this particular type of application, where only limited temperature swings are permissible, the volume of water or masonry necessary is very great - in the order of 10 times that required when PCM's are used.


Automatic 3 Way Valve
Collectors
Air Cooled
Heat Exchanger
To Use
High
Temperature
Controller
Sensor
Thermal
Storage
Return or Make-Up

Automatic Air Cooled Heat Dumping Circuit

FIGURE 6.1.4



Heat Storage In Phase Change Materials

FIGURE 6.2.1

Desirable characteristics for PCM's include:

(1) appropriate melting point

(2) little or no subcooling required to initiate solidification

(3) high heat of transition

(4) chemical stability (i.e. no loss of storage capacity over time)

(5) low cost

(6) nonflammability and nontoxicity

(7) small volume change on transition

(8) high thermal conductivity

(9) low vapour pressure

Two general classes of PCM's of current interest for solar use are organic compounds such as paraffin, and salt hydrates such as Sodium Sulphate Decahydrate, commonly called Glauber Salts. Organic PCM's tend to be more stable than salt hydrates but they have broad melting ranges which spread the heat of transition out over large temperature ranges. Other limitations of organic PCM's include low thermal conductivity, high flammability and high volume changes during melting. In salt hydrates, hydration and dehydration occur rather than simple freezing and melting. Salt hydrates are classified according to their phase change properties as (1) congruent (the solid and liquid phase composition have the same ratio of salt to bound water (simple freeze-melt)), (2) semi-congruent (the material has two or more hydrate forms with different melting points) and (3) incongruent (the solid and liquid phases have different ratios of salt to bound water which results in an insoluble anhydrous salt precipitate which, if allowed to settle out, results in reduced thermal storage capacities). Congruent salts with desirable characteristics are too expensive for most solar storage applications. The less expensive incongruent and semi-congruent salts have been the object of most research with most efforts directed at overcoming their problems.

The problems of getting the liquid solution to crystalize without subcooling has been overcome by using nucleating agents and by the use of a cool surface or "cold finger" in the PCM which promotes nucleation.

The settling out of precipitates in incongruent salts can be reduced in one of three ways. The salt hydrates in storage may be mechanically mixed, the crystals may be suspended in peat moss by the addition of thickeners or the material may be packaged in very thin horizontal layers (which reduces settling distances).

Containers housing PCM's which permit thermal expansion, are corrosion resistant, inhibit settling and encourage heat transfer and nucleation, can cost more than the PCM's they contain. To date, cost is the major drawback to the widespread use of PCM's in thermal energy storage systems.

TABLE 6.2.1

CHARACTERISTICS OF PHASE CHANGE MATERIALS

Name	Melting point ° C	Heat of Fusion	
		KJ/Kg	KJ/ℓ
Hydrated Inorganic Salts			
Sodium chromate	19.9	181.8	275
Manganese nitrate	26.3	140.1	245
Ortho Phosphoric acid	29.5	142.8	261
Lithium nitrate	29.9	294.5	677
Calcium chloride	30.2	169.7	282
Glauber Salt	32.4	237.2	346
Disodium phosphate	34.6	280.6	430
Manganous nitrate	35.4	116.9	232
Zinc nitrate	36.2	130.1	267
Calcium nitrate	42.6	141.7	257
Thiosulfate (Hypo)	48.6	94.4	175
Nickelous nitrate	56.7	152.1	310
Cobaltous nitrate	57.1	126.1	309
Cadmium nitrate	59.5	106.0	259
Sulfur trioxide	62.2	317.7	623
Magnesium nitrate	83.4	159.5	349
Hydrazine hydrochloride	92.7	221.9	287
Magnesium chloride	117.8	168.6	335
Anhydrous Inorganic Salts			
Arsenic tribromide	31.9	37.1	
Meta Phosphoric Acid	42.6	107.1	
Phosphoric acid	70	156.3	
Antimony trichloride	73.3	55.7	
Antimony tribromide	96.4	38.5	
Aluminum bromide	97.9	42.2	
Ammonium acid sulfate	143.9	124.1	
Ammonium nitrate	169.4	63.5	
Potassium thiocyanate	176.7	111.3	
Waxes and Organic Solids			
Anthracine	96.1	104.8	130
Anthraquinone	285	157.2	224
Naphthaline	80	150.5	172
Naphthol	95	162.3	197
Bees wax	61.7	176.7	167
Stearic acid (tallow)	76.1	198	167
Amorphous paraffin wax	74.4	229.6	182

6.3 CONTROLS

The controls in a solar energy system must operate the system so that it automatically collects and stores solar energy when it is available, and co-ordinates the application of this stored energy with the requirements of the energy user. Energy collection should be maximized, while electrical energy consumption should be minimized. In the event of a controls failure, the system should be designed to "fail safe". All this should be done simply, logically, dependably and cheaply.

The three basic components (Figure 6.3.1) of a control system are sensors, comparators and output devices. The sensors (usually temperature sensors) detect or measure variables at pertinent locations in the system. The comparators compare the conditions indicated by the sensors to some preset value or to the value measured by another sensor. This comparison is used to activate the output devices according to a prearranged plan. The output devices are mechanical components such as automatic valves, motor starters on fans or pumps and variable speed motor controllers.

The simplest and cheapest controllers are on-off units. Modulating controllers are more complex and costly, but lend themselves to more efficient system operation. However, the increased effectiveness is rarely worth the increased cost.

Controls are easiest understood if they are considered in sections. Usually a control system can be broken into a number of logical and separate segments. For example, the system shown in Figure 6.3.2 can be separated into three sections:

(1) collector to storage loop control.
(2) storage to preheat loop control.
(3) storage to space heating loop control.

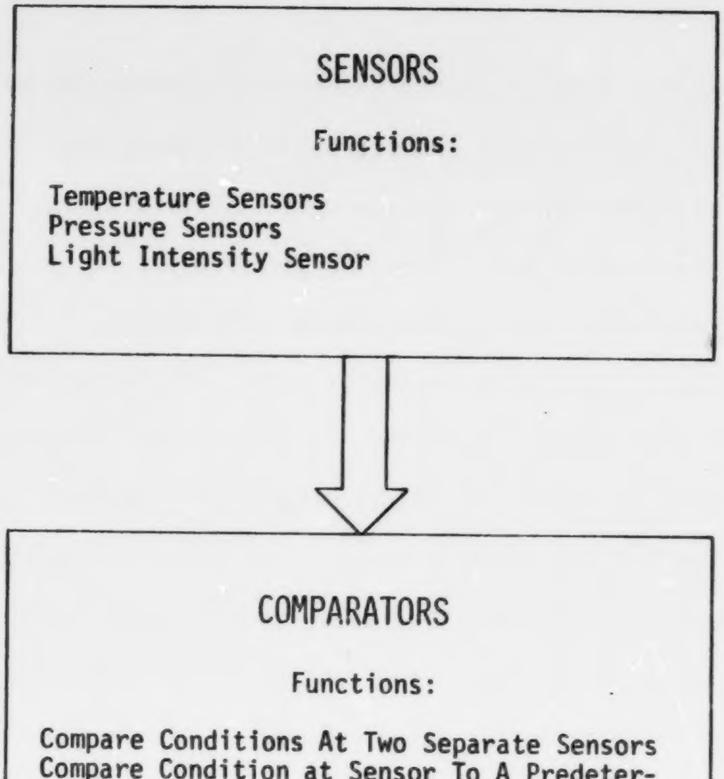
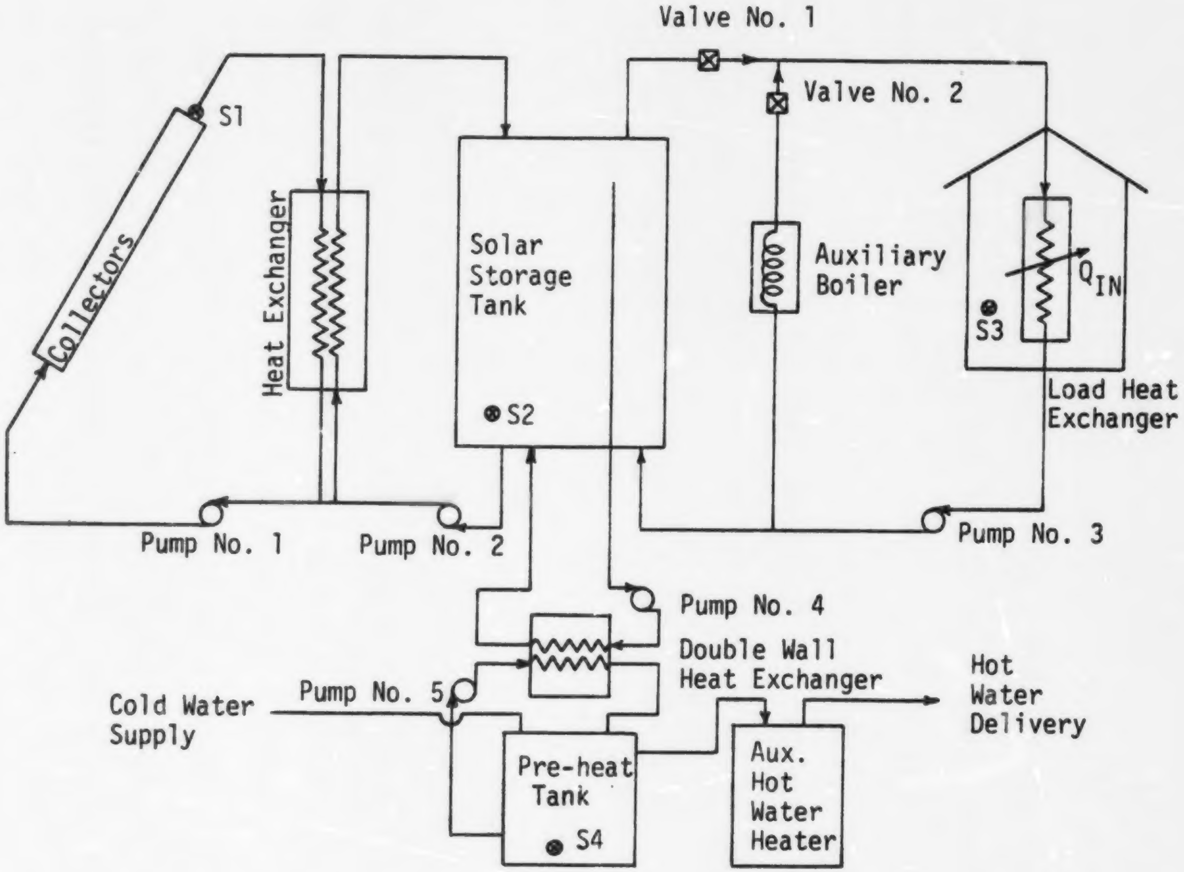

SENSORS
Functions:
Temperature Sensors
Pressure Sensors
Light Intensity Sensor
COMPARATORS
Functions:
Compare Conditions At Two Separate Sensors
Compare Condition at Sensor To A Predetermined Value
OUTPUT DEVICE
Functions:
Motor Starter
Solenoid Drive On Valve Or Damper

FIGURE 6.3.1


Valve No. 1
Valve No. 2
S1
Collectors
Heat Exchanger
Solar
Storage
Tank
S2
Auxiliary
Boiler
S3
Q_{IN}
Load Heat
Exchanger
Pump No. 1
Pump No. 2
Pump No. 3
Pump No. 4
Double Wall
Heat Exchanger
Cold Water
Supply
Pump No. 5
Pre-heat
Tank
S4
Aux.
Hot
Water
Heater
Hot
Water
Delivery

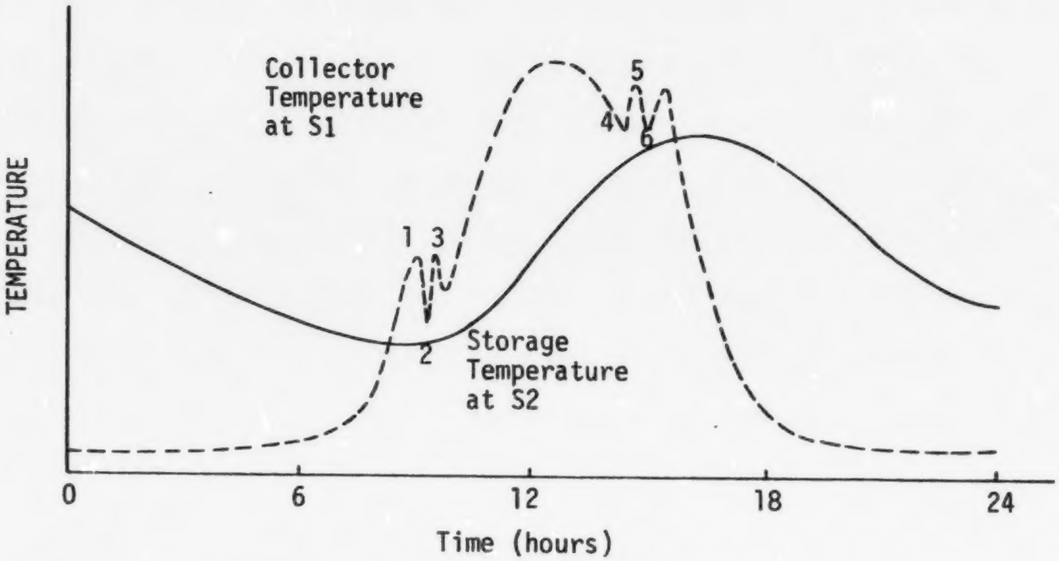
Sensor Locations in a Typical Liquid-Heating System

FIGURE 6.3.2

6.3.1 Collector To Storage Controls

When the solar collectors receive incident solar radiation, the temperature of the absorber plate rises, warming the transfer media. The collector sensor (S1 in Figure 6.3.2) senses this increasing temperature. A comparator compares the temperature of the solar collector and the thermal storage. When the temperature sensed at S1 is above that in storage at S2 by a predetermined amount, the circulation pumps (or fans) in the collector-storage loop are turned on. When the temperature difference falls to some lesser amount, the pumps turn off. The temperature differential necessary for pumps to start is typically 10°C, and the pumps cut off at a 2°C temperature differential. This deadband should be large enough to reduce on-off cycling of pump (or fan) motors, while keeping the starting temperature differential low enough so as not to significantly degrade the system efficiency. Figure 6.3.3 is an example of collector temperature profiles on a sunny day, which also indicates pump operation.

The heat transfer fluid is taken from the bottom of storage (as this is the coldest part of storage) and returned (after being heated) to the top of storage.This enhances thermal stratification in the storage and,because the coldest heat transfer fluid is being heated, collector efficiency should be maximized. The cut out temperature differential, typically 2° C, should reflect this thermal stratification and the temperature drop in the piping and ducts from the top of the collector to the storage inlet.



Typical Temperature Profiles of Collector and Storage Liquids in a Liquid-Heating System

FIGURE 6.3.3

6.3.2 Preheat Controls

Preheated domestic hot water may be taken directly from storage, or a separate preheat loop may be utilized (Figure 6.3.2).This separate preheat loop operates on the same logic as the collector-storage loop: taking hot fluid from the top of the solar storage tank, passing it through a heat exchanger and returning it to the bottom of the storage tank while cool water is taken from the bottom of the preheat tank, warmed in the heat exchanger, and returned to the top of the preheat tank. The temperature in the preheat water tank is compared to that of the solar storage to determine when the circulation pumps should operate. Preheated water is supplied to the regular water heater where it is reheated for use as necessary.

6.3.3 Space Heating Controls

Solar space heating controls must be integrated with the conventional space heating control system. One method of achieving this is by using a two stage thermostat. When the room temperature falls enough for the first stage to be engaged, heat from the solar storage is circulated through the heating system in an effort to meet the heat loads. If it cannot meet space heating loads, temperatures continue to fall until the second stage in the thermostat makes contact. When this happens, the controls function to bypass the solar space heating system and start the conventional heating system. (In the system shown in Figure 6.3.2, valve 1 closes and valve 2 opens, so that circulation is through the auxiliary space heater which has been activated to meet the space loads). When the system is operated to minimize cycling of control functions, significant temperature swings (5° C) occur in the space. Narrowing the operating bands to reduce the temperature swings promotes control cycling. One method to reduce room temperature swings is by limiting the use of storage to those times when it is above say 40° C or 50° C. When it is below this temperature, the auxiliary heater would automatically start when the first stage makes contact.

Another method is to separate the conventional and solar heating systems completely, and have the solar heating controls set slightly higher than the conventional system controls. If a demand for heat cannot be met by the solar heating system, the space temperature will fall enough to start the conventional system, which will raise space temperatures to something less than the set point of the solar heating thermostat, before turning the conventional system off. The solar heating system will continue to operate until all available stored energy is used, or until this system has met the space heat load.

This approach is one frequently used with air collectors. Air supplied at say 25° C is useful in meeting space heating loads while water storage must be 35° C or warmer to provide useful heat using a fan coil. This operating temperature difference may result in better system efficiency using air collectors over liquid collectors with a similar efficiency curve. It is possible to reduce the useful storage temperature of liquid systems through the use of heat pumps, which are now widely available.

6.3.4 Controls Selection And Placement Considerations

The location of sensors in solar energy systems is important for efficient system operation. Collector temperature should be sensed in a collector outlet at the top of the collector array. If possible, the sensor should be immersed in the solar fluid. If it cannot be directly immersed, special care should be taken to ensure that the sensors will give an accurate indication of the temperature of the solar fluid in the solar panel or of the absorber plate. The sensor on the solar collector should be able to withstand stagnation temperatures without breaking down or becoming detached from its mount.

A controller which interrupts the power supply to the pumps when low solar fluid temperatures are sensed should be located in the solar fluid line from the collectors to the heat exchanger. This would prevent freezing of the heat exchanger in the event of a controls failure (which operated the pumps despite low solar fluid temperatures).

In storage tanks, sensors should be near the bottom to detect the coldest tank temperature because the bottom water is that which is heated, and thermal stratification can be very significant.

Temperature limiters should be located at the top of the storage so that the maximum temperatures are sensed.

Pressure limiters should be set to reflect their location in the loop and they should be attached to the most pressure sensitive system component.

A method of testing control functions should be incorporated into the controls design. This will permit the operator to more readily trouble shoot the system in the event of a system failure.

A variety of good quality solid state controls designed specifically for solar applications are now marketed. These are generally quite inexpensive compared to controls systems made by mixing and matching standard HVAC components in an effort to achieve the appropriate controls logic. In addition, the solar industry designed controllers are less complex to install and are usually equipped with additional useful features.

A number of publications concerning controls for solar energy systems are listed in the references at the end of this chapter.

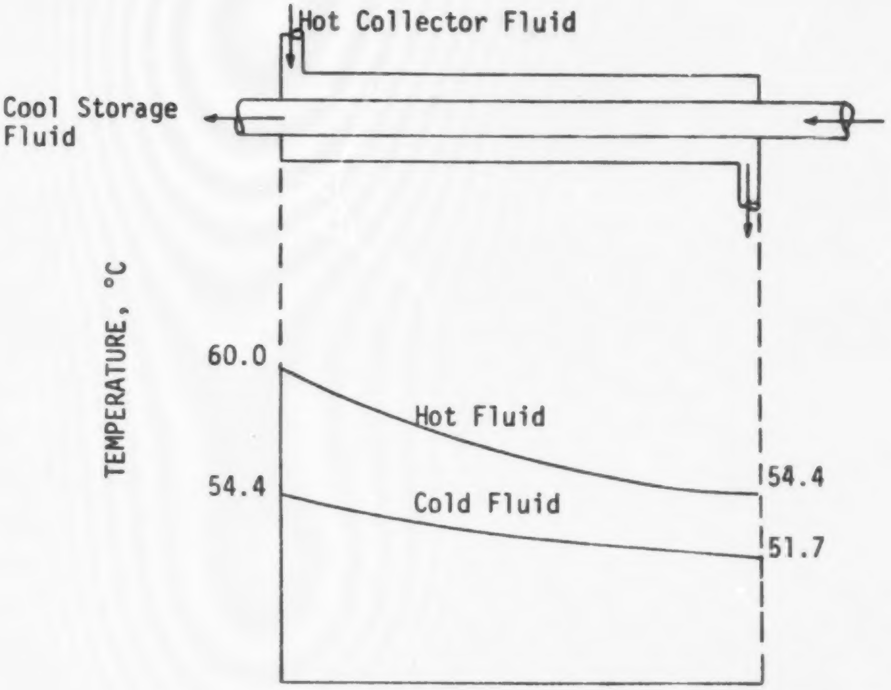
6.4 HEAT EXCHANGERS AND SOLAR FLUID

6.4.1 Introduction

Heat exchangers are available in a variety of configurations and construction materials, and can be used for a variety of applications. The selection of a heat exchanger configuration is dependent on the specific design criteria pertinent to the application at hand.

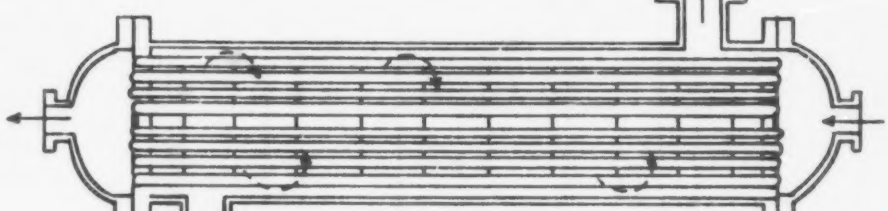
Counterflow heat exchangers, those in which the cold fluid flows in the opposite direction to the hot fluid (Figure 6.4.1), minimize the temperature differences between the outgoing hot (solar) fluid and the incoming cold (storage) fluid. Counterflow tube and shell multiple pass heat exchangers (Figure 6.4.2) are a compact and effective design suitable for use in solar energy systems. If the heat exchanger is used to heat domestic service water, double-walled heat exchangers will eliminate the possibility of solar fluid contaminating the domestic water and may be required by the plumbing code authority. A number of double-walled heat exchanger configurations are shown in Figure 6.4.3. There is not yet a national policy on this, and requirements vary from jurisdiction to jurisdiction. One standard for selecting the type of heat exchanger involves rating solar fluids according to their hazard (toxicity), and the possibility of a solar fluid leak going undetected. The combination of these hazards determines the type of heat exchanger that must be utilized for domestic water heating applications. Tables 6.4.1 through 6.4.4 highlight this approach using the guidelines set out in NRC's "Potable Water Protection in Domestic Hot Water Heating System" December, 1979, Contract #079-010.

The following sections outline considerations which can affect the safety of solar DHW heating systems.


Hot Collector Fluid
Cool Storage Fluid
TEMPERATURE, °C
60.0
Hot Fluid
54.4
Cold Fluid
54.4
51.7

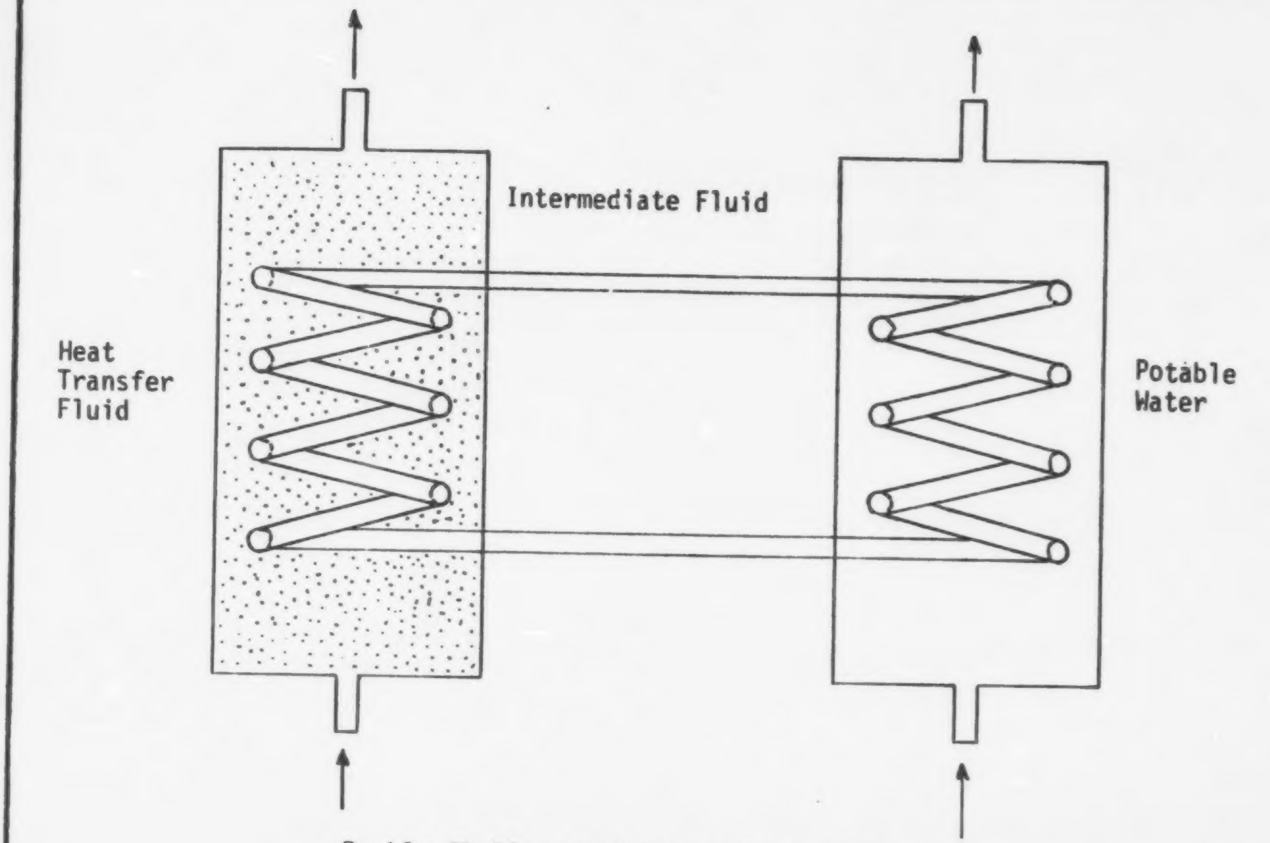
Single-Pass Counterflow Shell-and-Tube Heat Exchanger

FIGURE 6.4.1

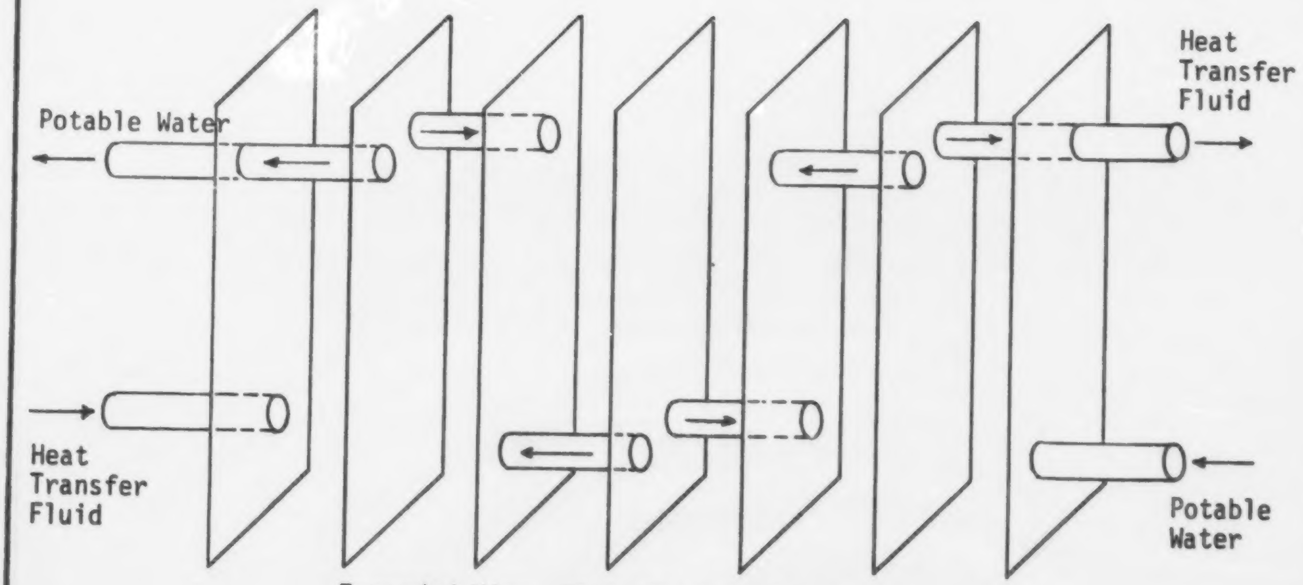


Multiple Tube Heat Exchanger

FIGURE 6.4.2

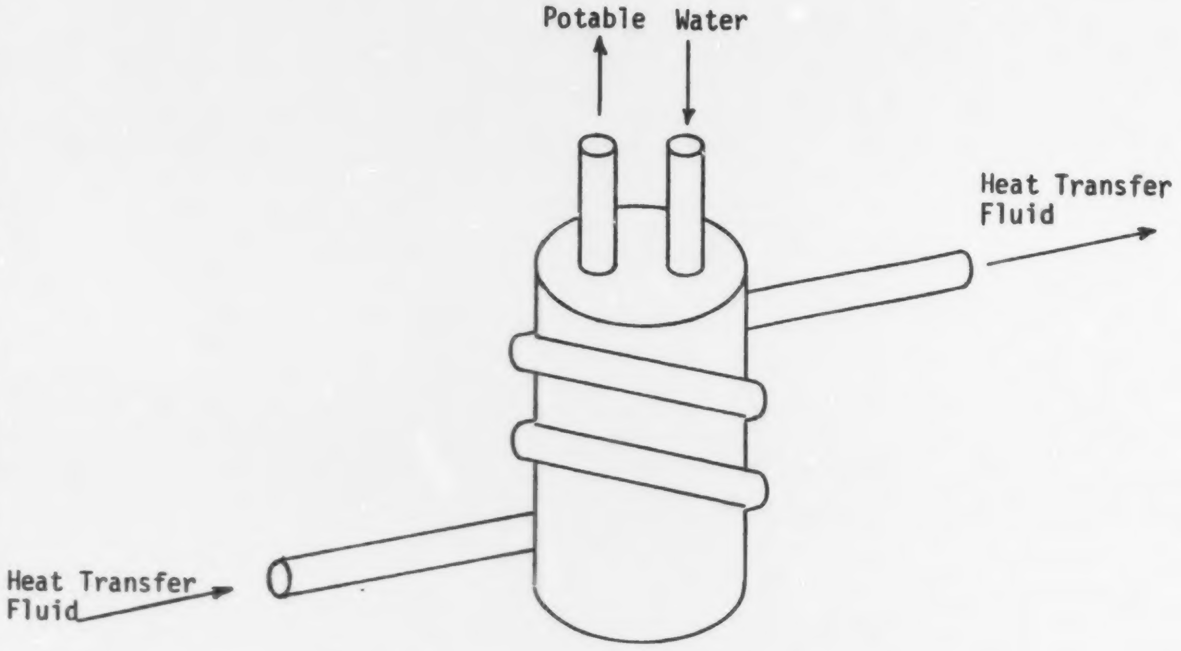

Intermediate Fluid
Heat Transfer Fluid
Potable Water

Double Shell and Tube Heat Exchanger

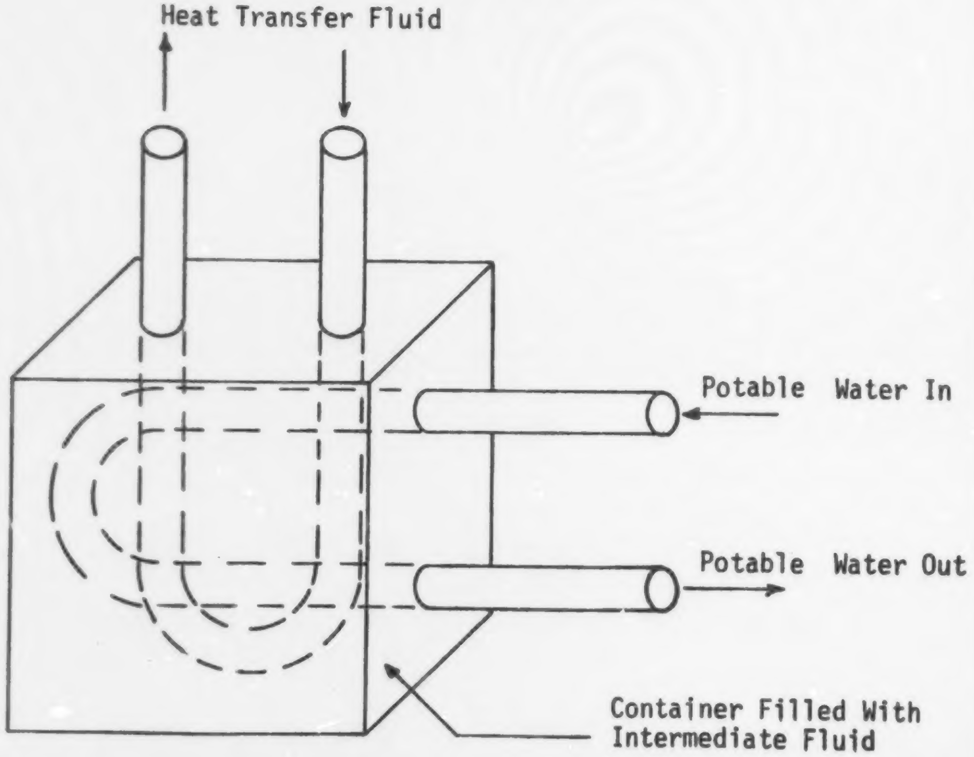

Potable Water
Heat Transfer Fluid
Heat Transfer Fluid
Potable Water

Expanded View Plate Heat Exchanger

FIGURE 6.4.3


Potable Water
Heat Transfer Fluid
Heat Transfer Fluid

Tube-On-Shell Heat Exchanger (Intimate Contact)


Heat Transfer Fluid
Potable Water In
Potable Water Out
Container Filled With Intermediate Fluid

Shell and Double Tube Heat Exchanger

FIGURE 6.4.4

TABLE 6.4.1

HEAT TRANSFER LIQUIDS AND HEAT SINK LIQUIDS CLASSIFICATIONS

CLASS A.a.	WATER SUITABLE FOR DRINKING.
CLASS A.b.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 1 AND CONTAINING A SUITABLE INDICATOR.
CLASS B.a.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 1.
CLASS B.b.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 2 CONTAINING A SUITABLE INDICATOR.
CLASS C.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 2.
CLASS D.	ANY LIQUID WITH A GOSSELIN TOXICITY RATING 3.

LIQUIDS WITH A GOSSELIN TOXICITY RATING OF GREATER THAN 3 SHALL NOT BE USED WITH ANY POTABLE WATER SYSTEM.

TABLE 6.4.2

LIQUID-TO-LIQUID HEAT EXCHANGER CLASSIFICATIONS

CLASS 1.	SINGLE WALL HEAT EXCHANGERS.
CLASS 2.a.	SINGLE WALL HEAT EXCHANGERS WITH AN ADDITIONAL LEVEL OF PROTECTION.
CLASS 2.b.	DOUBLE WALL HEAT EXCHANGERS.
CLASS 3.	DOUBLE WALL HEAT EXCHANGERS WITH AN ADDITIONAL LEVEL OF PROTECTION.
CLASS 4.	DOUBLE WALL HEAT EXCHANGERS WITH A VISUAL MEANS OF LEAK DETECTION.

TABLE 6.4.3

GOSSELIN TOXICITY RATINGS OF SOME COMMONLY USED SOLAR FLUIDS

RATING	MEANING	EXAMPLES
0	NONTOXIC	POTABLE WATER
1	PRACTICALLY NONTOXIC	PROPYLENE GLYCOL
2	SLIGHTLY TOXIC	ETHYLENE GLYCOL ETHYL ALCOHOL (BORDERLINE 2TO3)
3	MODERATELY TOXIC	PHENYL ETHER-DYPHENYL MIXTURE (DOWNTHERM A*)
4	VERY TOXIC	
5	EXTREMELY TOXIC	

---

* COMMON TRADE NAME.

TABLE 6.4.4

PERMISSIBLE HEAT EXCHANGER - HEAT TRANSFER FLUID COMBINATIONS

HEAT TRANSFER FLUID CLASS	HEAT EXCHANGER CLASS			
	1	2	3	4
A	P	P	P	P
B	N	P	P	P
C	N	N	P	P
D	N	N	N	P

N - NOT PERMITTED

P - PERMITTED

6.4.2 Reducing The Possibility Of Leakage

Careful component design and selection will reduce the potential for component failure which will result in leakage of solar fluid. Component failure may be a result of mechanical failures such as faulty welding or brazing at heat exchanger seams and joints, gasket failures or component failure due to excessive thermal shock. Corrosion in systems using reactive metals can be severe. The pH of the solar fluid, temperature, dissolved oxygen levels, the concentration of dissolved chemicals, and junctions between dissimilar metals can all rapidly accelerate corrosion. High flow velocities through system components will result in erosion, especially where flow directions change, as they do in heat exchanger U tubes. If careful attention is paid to the selection of the materials used in the system, and the design and construction of the components, the risk of leakage can be minimized.

6.4.3 Reducing Potential For Leakage Between Solar Fluid And DHW Supply

The possibility of leakage occurring between the solar fluid and the water being heated can be most effectively eliminated by reducing the potential points of cross-flow. Typically, the only place where cross-contamination can occur in a dual fluid solar energy system is in the heat exchanger.

Double-walled heat exchangers, if correctly designed and installed, can virtually eliminate the possibility of cross-contamination. There is, however, a cost/performance penalty of using double-walled heat exchangers instead of single-walled units. Experience indicates that performance information on double-walled heat exchangers is not readily available, and selection is limited.

An alternative to eliminating the potential for cross-contamination due to leakage is to attempt to restrict the direction of any leakage. By designing the solar energy system to operate with lower pressures in the solar fluid loop than in the domestic water loop, any leakage that does occur should occur from the domestic water into the solar fluid. The installation of a properly sized expansion tank and properly set pressure limiting devices will ensure that the pressure of the solar fluid is lower than operating pressures in the water system.

A failure to maintain adequate pressure in the domestic water loop could lead to contamination, unless some measure to prevent it is provided. One safeguard is a differential pressure trip which automatically drains the solar fluid loop if its pressure exceeds that on the water side. A second is to let any leak cause a blowdown, as described in the following section.

6.4.4 Detection

If a leak occurs, the ease of its detection is important. Leakage across the heat exchanger would occur from the higher pressure fluid to the lower pressure fluid. Normally the domestic water loop pressure would exceed the solar fluid pressure. Therefore, leakage would be into the solar fluid, raising the pressure in the solar fluid loop and diluting the solar fluid. This would show up as higher than normal pressure in the solar loop, which should result in solar fluid blowing down. Reduced solar fluid concentrations may make the collectors susceptible to freezing damage.

For leakage to occur into the domestic water, the pressure in the solar fluid loop must exceed the pressure in the water loop. This could occur if water pressures are drastically reduced or if component failures cause excessive pressure buildups in the solar fluid loop.

The inclusion of a dye or a mercaptan in the solar fluid would stain or odour the water if solar fluid leaks into the water supply. Some commercially available solar fluids contain a concentrated food dye. In any case, users of the water should be informed as to the warning signs.

6.4.5 Solar Fluid Toxicity

The solar system designer has some latitude in specifying solar fluids for a project. Although nontoxic fluids are available, heat transfer fluids should always be regarded as nonpotable. A nontoxic solar fluid may become toxic due to chemical additives and inhibitors, from impurities picked up or dissolved from the solar loop or as a result of decomposition due to impurities, aging, or high operating temperatures. Further, there is always a possibility that a nontoxic fluid will be replaced with a toxic one in the future. The fill point of the solar loop should be labelled, clearly specifying the solar fluid to be used in the system and the expected life of the solar fluid.

The use of the solar heated domestic water is significant in assessing the potential hazard. Normally, domestic hot water is not consumed, but is used for washing and bathing purposes. This greatly reduces the danger as the allowable concentration level for skin contact is generally orders of magnitude greater than allowable concentration levels in drinking water. Obviously if only nontoxic solar fluids are used, the toxicity hazard does not exist. Nontoxic or low toxicity solar fluids such as propylene glycol, silicone fluids, or paraffinic oils are recommended for potable water systems if the possibility of water-solar fluid cross-contamination exists.

6.5 SIZING A HEAT EXCHANGER FOR A DUAL FLUID SOLAR ENERGY SYSTEM

The purpose of the heat exchanger is to transfer the heat energy absorbed by the solar fluid in the collectors to the thermal storage. If the heat exchanger is too small, insufficient heat will be removed from the solar collector system and the leaving exchanger liquid temperature will be high. This will cause a high entering liquid temperature into the solar collector. High liquid temperature in the collector will reduce collection efficiency. Given an adequately-sized heat exchanger, the liquid temperature rise in the solar system will be moderate, leading to operation of the collection system at maximum obtainable efficiency.

The design of a glycol collector subsystem must consider a "trade-off" between the solar collector area and the glycol to water heat exchanger surface area. For the same rate of solar heat input into storage, an undersized heat exchanger will require a much larger collector surface area as compared with an adequately-sized heat exchanger. Since collector surface area is very costly compared with heat exchanger surface area, the trade-off should be towards generous sizing of the exchanger in order to maintain collection efficiency, thus reducing collector area for equal solar heat input into storage.

The following section does not attempt to explain heat transfer in such detail as to allow the solar system designer to design the required glycol to water heat exchanger. The subject is too complex for treatment here. However, it is intended to provide an overview of heat exchanger sizing in a glycol-cooled collector system so that the solar designer can exercise judgement in terms of exchanger specification, leaving the final design and sizing to experts in this field.

Liquid to liquid heat exchangers used in solar water heating systems experience different thermal operating conditions than do heat exchangers used in boiler water heating systems. Manufacturers of heat exchangers usually use computer programs to select heat exchangers for customers.These programs make the selection based upon one set of (constant) conditions, that the heat exchanger will experience, as specified by the customer. However, in hydronic solar systems, the operating conditions of the heat exchanger are not constant. The diurnal climatic fluctuations and load variations cause significant changes in the hot fluid temperature throughout the day. If a heat transfer fluid is used that has a viscosity that changes significantly over the temperature range typically experienced in a flat plate solar collector, 15 - 80°C (59 - 180°F), as the viscosities of solutions of ethylene and propylene glycol do, special care must be taken when selecting the heat exchanger design parameter values.

Changes in viscosity affect heat transfer in two ways:

1. Increased viscosity reduces fluid flow, thereby decreasing the turbulence of the fluid, and lowering the conductance of a heat exchanger.
2. Increased viscosity causes a "thickening" of the laminar flow boundary layer at the wall of a heat exchanger also lowering the conductance of a heat exchanger.

The effects of increased viscosity on heat exchanger conductance can be countered by increasing the flow rates of the fluids. However, this solution is not always practical because of the increased pumping costs.

In summary, a heat exchanger for a solar water heating system, chosen by using a heat exchanger manufacturer's computer program, will be incorrectly sized unless the values of the thermal sizing parameters are consistent with the operating conditions actually experienced by the heat exchanger.

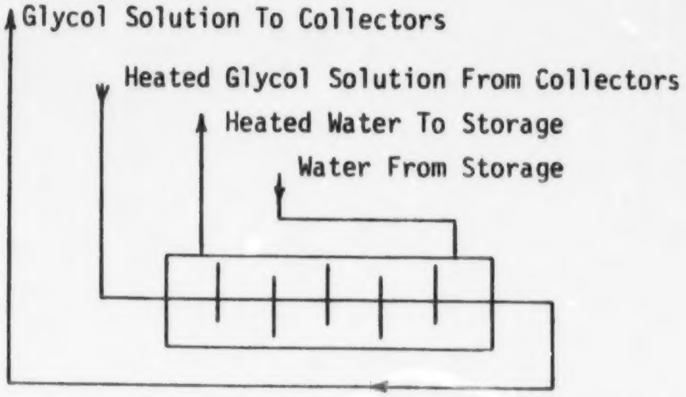
Three basic types of glycol to water heat exchanger units are described for use with glycol-cooled collector subsystems. They are illustrated in Figure 6.5.1.

The counter flow straight tube heat exchanger (A in Figure 6.5.1) will be used to describe approach temperature difference as this factor influences heat exchanger sizing and consequent collector operating efficiency.

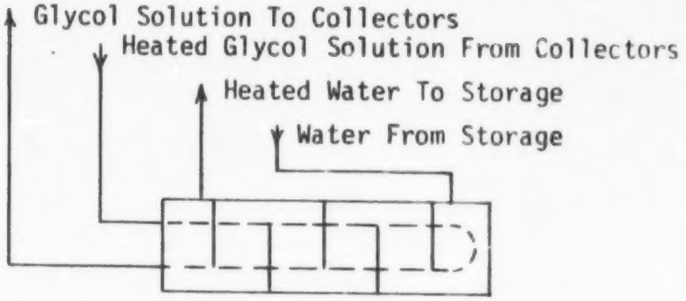
Approach temperature difference for a heat exchanger is the difference in temperature between the incoming heating fluid and the leaving heating fluid. Figure 6.5.2 illustrates the approach temperature difference (ΔT) for a glycol to water solar heat exchanger. The exchanger is designed for a 10°F (5.5°C) "approach".

The approach ΔT used for design of the heat exchanger has great significance for glycol-cooled collector systems because of its effect on the solar panel temperature required to meet a stated heat exchanger load. The higher the design approach ΔT, the higher the mean solar panel liquid temperature required to meet the load.

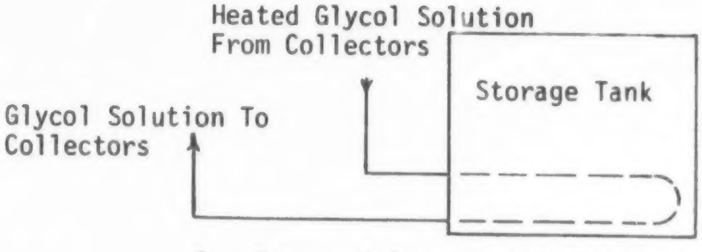
Figure 6.5.3 illustrates the effect of heat exchanger design approach ΔT on required solar collector liquid temperatures for a fixed load.


Glycol Solution To Collectors
Heated Glycol Solution From Collectors
Heated Water To Storage
Water From Storage

A: Counterflow Straight Tube Heat Exchanger


Glycol Solution To Collectors
Heated Glycol Solution From Collectors
Heated Water To Storage
Water From Storage

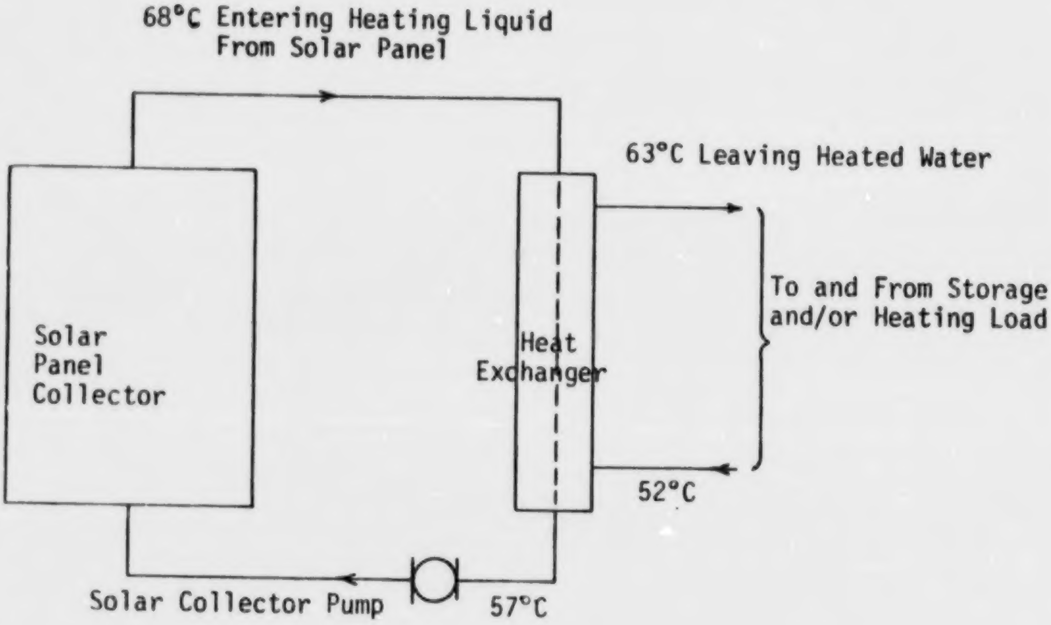
B: "U" Tube Heat Exchanger


Heated Glycol Solution
From Collectors
Storage Tank
Glycol Solution To
Collectors

C: Immersed Storage Tank Heat Exchanger

Heat Exchangers Used for Glycol Solution
Filled Solar Collector Systems

FIGURE 6.5.1


68°C Entering Heating Liquid
From Solar Panel
63°C Leaving Heated Water
To and From Storage
and/or Heating Load
Solar
Panel
Collector
Heat
Exchanger
52°C
Solar Collector Pump
57°C

Heat Exchanger Designed for 5° Approach (68°-63°-5°)

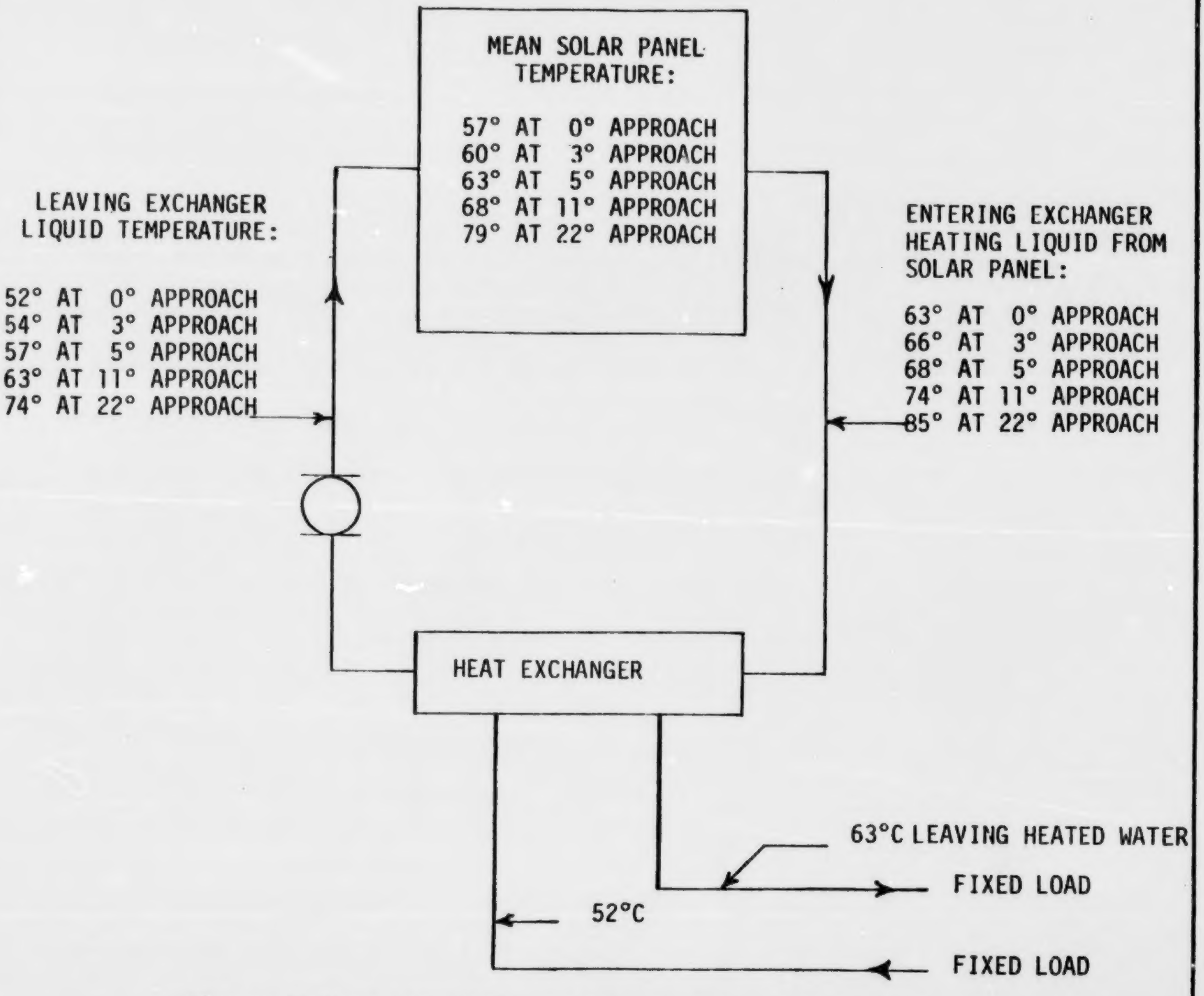
FIGURE 6.5.2

The 0° approach shown in Figure 6.5.3 and in Table 6.5.1 refers to a collector system which has no heat exchanger (water drain back system) and serves as a base for evaluation of the use of approach ΔT for the specification of solar glycol to water exchangers.

The glycol to water heat exchanger used in solar systems should be base sized for a 2.8°C to 8.3°C (5°F to 15°F) approach, with 5.5°C (10°F) as the recommended value. The reason is that exchangers sized to less than a 2.8°C approach will provide very little increased collector panel performance for a comparatively large increase in exchanger size and cost. Heat exchangers sized for greater than an 8°C approach will reduce panel performance excessively (relative to the decreased cost of the exchanger).

It should be noted that approach ΔT is a rough comparative measure of heat exchanger size. A heat exchanger selected for a 22°C (40°F) approach will be about 1/4 the size of an exchanger selected for a 4.5°C (10°F) approach. The difference in cost will generally not be proportionate. Considering the relatively high installed cost of the solar collector panels, it would be most unwise to sacrifice panel performance for the relatively small decrease in exchanger cost established by design to a large approach ΔT.

The effectiveness of a heat exchanger, ε, compares the actual heat transfer rate, to the thermodynamically limited maximum heat transfer rate, as would be realized only in a counterflow heat exchanger of infinite heat transfer area, and may be determined by the following formulae:


MEAN SOLAR PANEL TEMPERATURE:
57° AT 0° APPROACH
60° AT 3° APPROACH
63° AT 5° APPROACH
68° AT 11° APPROACH
79° AT 22° APPROACH
LEAVING EXCHANGER LIQUID TEMPERATURE:
52° AT 0° APPROACH
54° AT 3° APPROACH
57° AT 5° APPROACH
63° AT 11° APPROACH
74° AT 22° APPROACH
ENTERING EXCHANGER HEATING LIQUID FROM SOLAR PANEL:
63° AT 0° APPROACH
66° AT 3° APPROACH
68° AT 5° APPROACH
74° AT 11° APPROACH
85° AT 22° APPROACH
HEAT EXCHANGER
63°C LEAVING HEATED WATER
FIXED LOAD
52°C
FIXED LOAD

Change in Mean Liquid Temperature in Solar Collectors
with Change in Heat Exchanger Design Approach ΔT

FIGURE 6.5.3

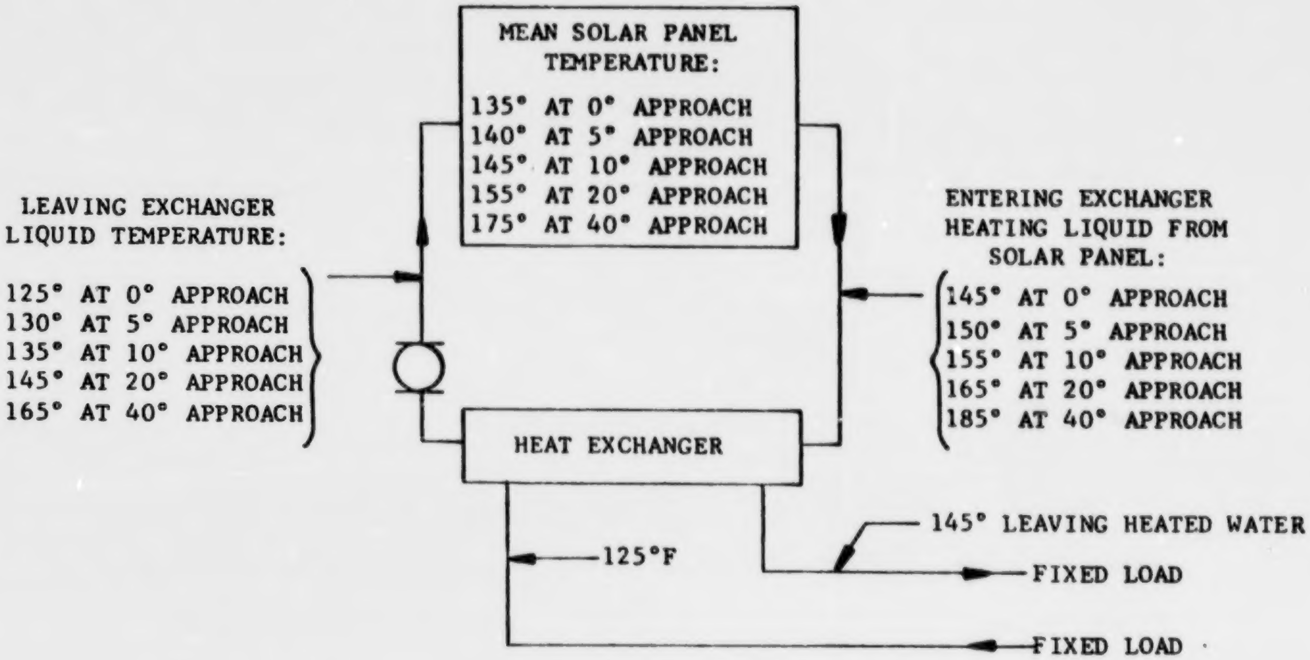

MEAN SOLAR PANEL TEMPERATURE:
135° AT 0° APPROACH
140° AT 5° APPROACH
145° AT 10° APPROACH
155° AT 20° APPROACH
175° AT 40° APPROACH
LEAVING EXCHANGER LIQUID TEMPERATURE:
125° AT 0° APPROACH
130° AT 5° APPROACH
135° AT 10° APPROACH
145° AT 20° APPROACH
165° AT 40° APPROACH
ENTERING EXCHANGER HEATING LIQUID FROM SOLAR PANEL:
145° AT 0° APPROACH
150° AT 5° APPROACH
155° AT 10° APPROACH
165° AT 20° APPROACH
185° AT 40° APPROACH
HEAT EXCHANGER
125°F
145° LEAVING HEATED WATER
FIXED LOAD
FIXED LOAD

Figure 6.5.3 Change in Mean Liquid Temperature in Solar Collectors with Change in Heat Exchanger Design Approach ΔT

The increase in mean solar panel temperature with increased heat exchanger approach temperatures will affect collection efficiency as follows for the collector described in Figure 3.2 and for 40° outdoor temperature.

TABLE 6.5.1

HEAT EXCHANGER APPROACH ΔT °F	MEAN PANEL TEMPERATURE °F	OUTDOOR AMBIENT °F	ΔT; MEAN PANEL MINUS OUTDOOR °F	COLLECTOR PANEL EFFICIENCY AT 250 B/H/FT^2 INSOL.
0° (Base)	135	40	95	33
5°	140	40	100	32.5
10°	145	40	105	32
20°	155	40	115	29
40°	175	40	135	24

$$\varepsilon = \frac{C_h\,(T_{\text{hot in}} - T_{\text{hot out}})}{C_{\min}\,(T_{\text{hot in}} - T_{\text{cold in}})} \qquad \text{(Eqn 6.5.1)}$$

$$\varepsilon = \frac{C_c\,(T_{\text{cold out}} - T_{\text{cold in}})}{C_{\min}\,(T_{\text{hot in}} - T_{\text{cold in}})} \qquad \text{(Eqn 6.5.2)}$$

Where C min is the smaller of Cc and Ch

C_c = the capacitance of the cold fluid J/s C° (BTU/hr. F°)

C_h = the capacitance of the hot fluid J/s C° (BTU/hr. F°)

The solar system designer will have to specify selection conditions (type of fluids, approach ΔT, expected liquid temperature and flow rates, allowable head loss, fouling factors, etc.) in order to allow the heat transfer specialist to select the lowest cost, most effective heat exchanger for the system.

When shell and tube exchangers are used, the solar system designer should allow the specialist the following options in order to obtain best performance at lowest cost.

1. The heat exchanger may be either "U" tube or counter flow straight tube. (See Figure 6.5.1)
2. The heat exchanger may be either single or multiple units or passes.
3. Glycol solution flow may be either through the tube or through the shell.
4. A maximum approach temperature of 5.5°C is recommended. While the specified approach ΔT should not be exceeded, a lower ΔT should be allowed. This will often permit the use of "off the shelf" exchangers of larger size than actually needed, but which have a lower cost than the units specifically manufactured to meet exacting specifications.
5. The maximum heat exchanger length may be specified by the specialist. Sufficient room must be allowed to permit removal of tubes from shell of heat exchanger.

6. The design temperature for the heat exchanger may be specified by the specialist. As the heat transfer characteristics of glycol solutions vary greatly with temperature, this is an important variable. The recommended design temperature is 40°C (or 104°F).

7. The specialist may specify fluid flow rates that are as high as practical to maximize heat transfer rates.

6.6 PUMPS AND PIPING

Centrifugal pumps are the best choice for pumping clean fluids against moderate pressures. They are readily available for a wide variety of capacities and pressure ranges at reasonable cost.

Pumps for solar fluids require special consideration in their selection. Mechanical seals which work for water may not be effective in preventing solar fluids from leaking. Pump specifications should make reference to the solar fluid being pumped, so that the supplier can select appropriate pump seals.

The viscosity of most solar fluids changes greatly with temperature, affecting flow rates and line pressures. The pump selected must be capable of operating without overloading over the full range of operating viscosities, and must move the required amount of solar fluid in the normal operating ranges.

Pump motors should be capable of handling slight overloads, but should not be excessively oversized. This is because electric motors under 50 kW require virtually same amount of power input, regardless of load in the normal operating range. A typical motor power-efficiency curve, illustrated in Figure 6.6.1, helps to demonstrate this concept. If the load on an electric motor is doubled from a to b, the efficiency increases from 40% to 70%, with an electrical energy consumption increase of only 14%.

To minimize heat and head losses, pipe runs should be as short as possible and all pipes, valves, tanks, heat exchangers, etc. should be

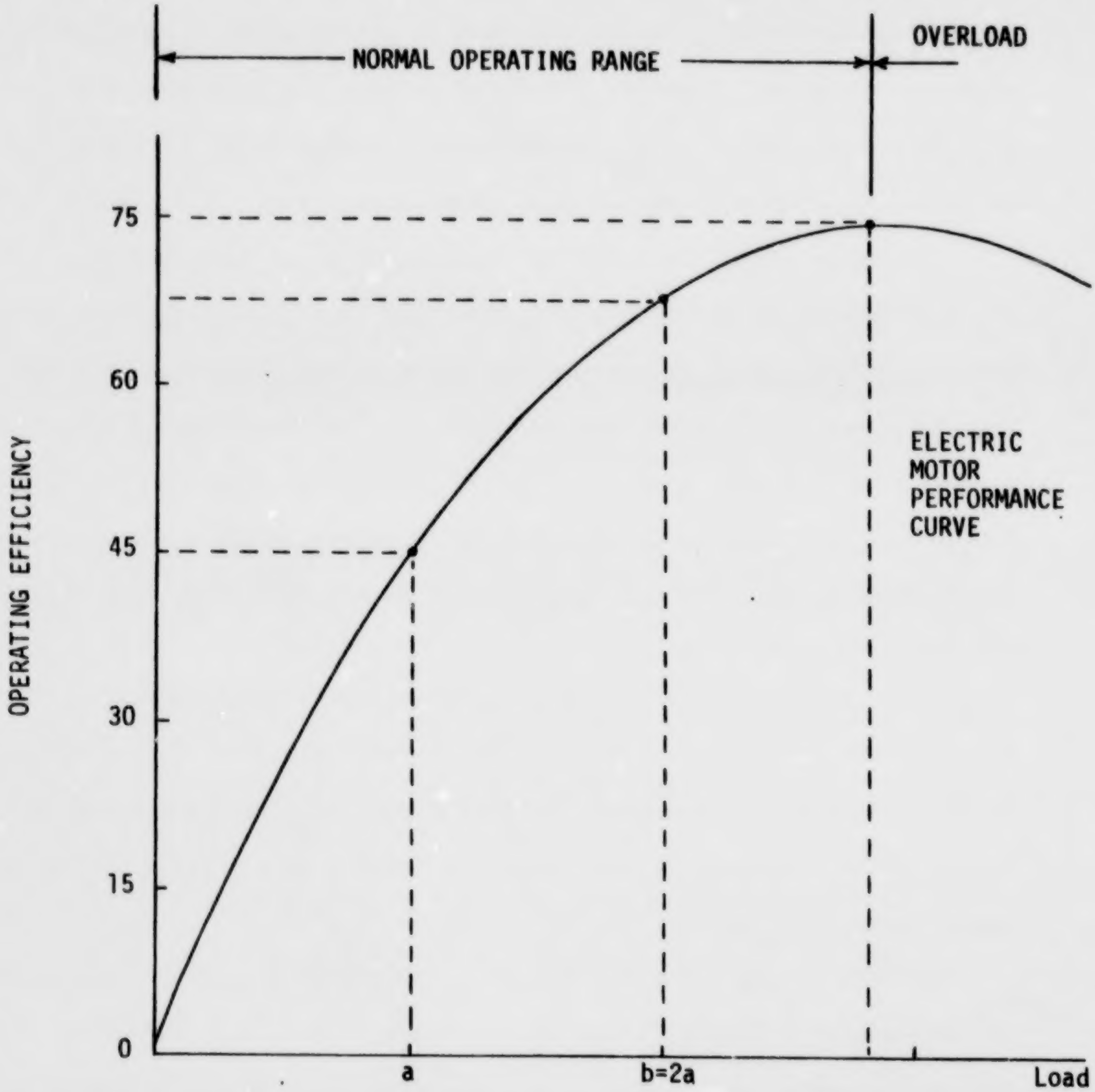

NORMAL OPERATING RANGE
OVERLOAD
75
60
45
30
15
0
OPERATING EFFICIENCY
ELECTRIC MOTOR PERFORMANCE CURVE
a
b=2a
Load

FIGURE 6.6.1

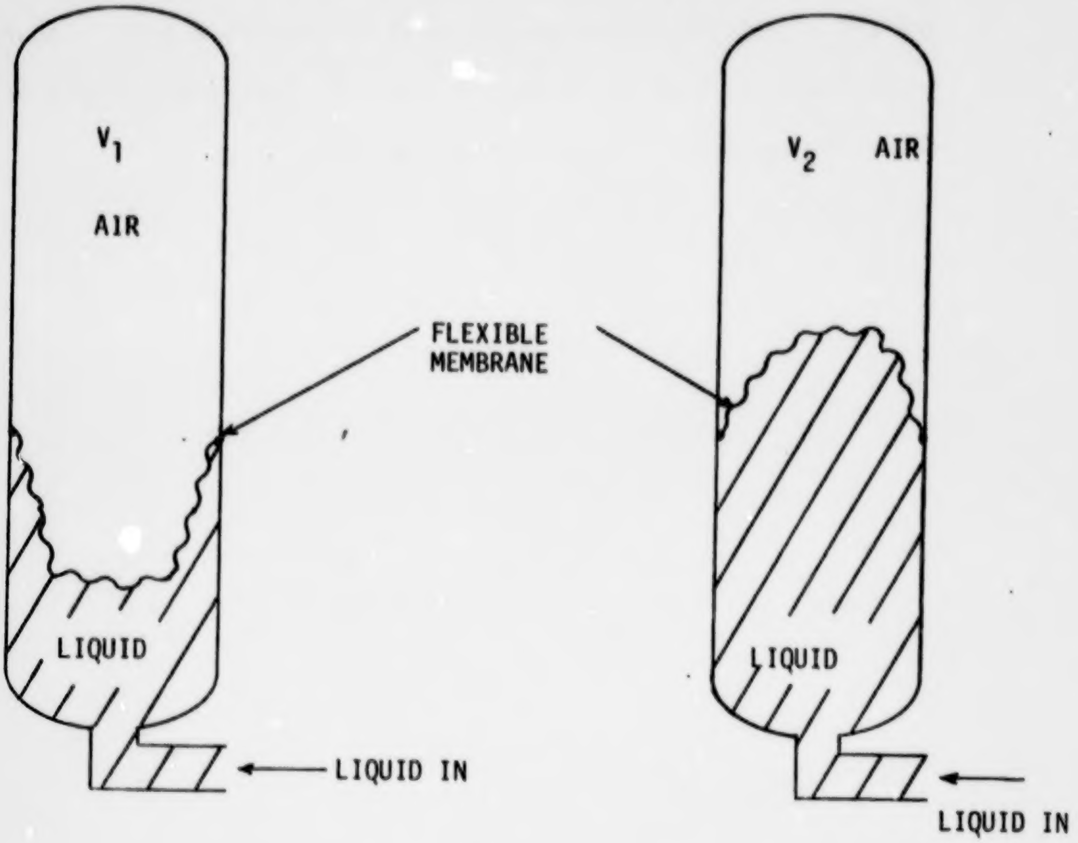
adequately insulated. The rule of thumb in sizing pipes is to select flow velocities of 1.5 meters per second or less. Higher velocities result in noise, erosion of piping and excessive head losses. Careful selection of other system components will reduce overall head losses.

Strainers should be installed upstream of pumps to reduce wear. Pressure gauges located around pumps are useful in quantifying flow through the pump when performance testing the system. They can also draw attention to unusual changes in pump performance. Similarly, temperature gauges located on the inlets and outlets of the heat exchanger are helpful in evaluating system performance and in selecting optimum temperature differences for the controls functions. There are PUSH guidelines regarding necessary monitoring systems for solar projects.

Solar fluid circulation loops experience very major temperature fluctuations, perhaps up to 200°C. The thermal expansion and contraction in piping and collector headers may be extreme, requiring the use of expansion loops, expansion compensators or flexible couplings in piping runs and between collectors.

Similarly, thermal expansion of the solar fluid must be facilitated. In unpressurized systems, freeboard or an overflow in the thermal storage tank may accommodate this expansion. In pressurized thermal storage, pressure relief valves and/or expansion tanks can absorb volume changes. Volume changes of 5% can occur in water storage fluctuating between 4°C and 100°C. Glycol solution fluctuating between 0°C and 150°C experiences volume changes of over 13%. When selecting an expansion tank for a closed loop system, the designer must know the volume change to be accommodated and the acceptable operating pressures in the fluid loop. Figure 6.6.2 shows a schematic of an expansion tank and shows the pressure volume relationships used in calculating tank size.

AIR VOLUME V = TANK VOLUME - LIQUID VOLUME


V_1
AIR
V_2 AIR
FLEXIBLE MEMBRANE
LIQUID
LIQUID
LIQUID IN
LIQUID IN

BUT

$V_2 = V_1$ - CHANGE IN LIQUID VOLUME

$$P_1V_1 = P_2 (V_1 - \Delta V)$$
$$P_2 \Delta V = P_2V_1 - P_1V_1$$
$$P_2 \Delta V = (P_2 - P_1) V_1$$
$$V_1 = \frac{P_2 \Delta V}{P_2 - P_1}$$

FIGURE 6.6.2

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CHAPTER 7

DURABILITY OF SOLAR COLLECTORS AND SYSTEMS

7.1 DURABILITY OF SOLAR COLLECTORS

Like all system components, solar collectors are subject to a variety of potential problems which can cause degradation of performance, reduced life and increased maintenance burdens. Some of the more common problem areas are discussed below. When evaluating collectors, the system designer should be very conscious of how manufacturers have handled these potential problems in their designs.

1. Glazing systems - A summary of properties of some common cover plate materials is given in Table 3.2.1. Glazing systems are subject to breakage from foreign object impact and thermal stresses, discolouration, embrittlement, loss of physical stability and abrasion. Breakage from foreign objects (vandalism) has not proved to be a major problem, even in public schools. Most breakage of glazing appears to be the result of excessive thermal stresses. For this reason, tempered or heat-strengthened glass is generally used where glass is specified. Plastics are less subject to breakage although Tedlar covers have been known to fail in extreme cold because they were overtightened on their frames. Discolouration of plastic glazings is caused by ultraviolet light reacting with polymer molecules. Polycarbonates such as Lexan, are particularly susceptible to this type of

damage. FRP (fiberglass) glazings generally have protective coatings or inhibitors added to provide UV protection. Fluorocarbons (PVF) such as Tedlar, are subject to embrittlement if not properly protected. Many plastics are also vulnerable to loss of physical stability under high temperature conditions. Some films, for example, can sag enough to actually touch the absorber plate. In very windy areas, abrasion due to wind-borne dust and dirt can also be a problem.

2. Absorber Plates - Certainly the greatest problem with absorber plates in liquid collectors is the possibility of corrosion. Corrosion in collectors and systems is discussed in more detail in Section 6.3. Since most absorber plates are metal they are also subject to problems with thermal expansion which, under extreme conditions, can cause buckling of up to 3 cm. Thermal expansion can also break soldered joints. Blockage of flow channels due to foreign objects is a remote possibility but can be handled with proper filtration.
3. Absorber coatings - Absorber coatings may be subject to degradation and discolouration due to extreme heat or to moisture. Selective surfaces are vulnerable to greying and bleaching. There is also the potential of deficiencies in the coating application. For example, thick coatings can crack or outgas under high temperature conditions. A summary of absorber coating properties is given in Table 3.2.2.

4. Insulation - Problems with insulation include outgassing from the binder or from foam insulation, moisture degradation and dimensional instability. Loose fill insulation will settle and is seldom used. Polyurethane (foamed in place) is unstable above 50° C and has caused absorber plate buckling.
5. Condensation - Condensation generally occurs when water vapour inside the collector box condenses on a cool cover plate during periods of low ambient temperatures. It can seriously degrade collector performance, ruin the absorber surface finish and cause or accelerate corrosion. Therefore all collectors should contain some type of condensate control mechanism. Three types of solutions are generally used: desiccants (usually located in breather hole), hermetic sealing of the collector (with the air inside being purged to remove any moisture) and weep holes and vents (which must be kept free and unclogged at all times).
6. Air Leakage - Proper sealing is of extreme importance to collector performance. Seals can dry out, embrittle or otherwise degrade in ultra-violet exposure. Butyl seals are very common, but experience indicates poor results; silicone seals give the best results but are very expensive. Leakage is primarily a problem with air collectors. Interestingly enough, it can actually improve system performance under certain conditions. For example, if the leakage is into the collector and is less than the fresh air requirements of the building (and is therefore required anyway), then the average operating temperature of the collector

will be lowered and hence the efficiency increased. For preheating systems, leakage is not a problem.

7.2 DURABILITY OF SOLAR ENERGY SYSTEMS

Many of the historical problems which have plagued solar energy systems have been the result of poor design and construction procedures which often ignore commonly accepted practices. With the industry now beginning to mature , many of these problems are being eliminated.

1. Transport systems - The most common problems are inadequate insulation in liquid systems and leakage in air systems. Since both are difficult to correct after the sytem has been installed, they should be carefully attended to prior to final systems commissioning. Insulation levels should be higher than normal to reflect the higher cost of heat from solar energy systems. Air systems should be carefully checked to seal all leaks before the insulation is applied. Leakage past outdoor air dampers has been known to freeze heat exchangers. Flexible plastic and rubber connectors often fail due to temperature, ozone, UV degradation or poor clamping.
2. Storage - The storage systems are generally trouble-free. There may be some biological growth in both air and liquid systems. Inadequate insulation and leakage in very large storage tanks has also been reported.

3. Controls - Controls are often too expensive, complicated and, as a result, trouble-prone. Attempts at oversophistication result in too many modes of operation with extensive numbers of devices to control. Improperly adjusted setpoints will cause cycling and result in unnecessary wear on the system. Temperature sensors, especially those in the collectors are sometimes improperly located with no provision for replacement. (Many solar collectors do not have provision for sensors.) Valves and dampers that close too slowly will waste energy, while solenoid valves that close too quickly cause water hammer. Thermosiphoning may result if reverse flow valves or dampers are not included. Monitoring equipment to diagnose faults or monitor performance is often lacking.
4. Installation - Ice dams on roofs can be caused by collector shading. Increased snow and wind loads on roofs can be created by a collector array and must be accounted for at the design stage. Note that a collector tilt angle of approximately 65° is required to shed snow. Problems may also arise from inadequate room for snow run-off or inadequate sealing of penetrations in the building membrane. The design should also leave room for movement of leaders and modules. Cranes are often required for collector installation. Mounting structures can be very expensive, and the use of a vertical wall for the collectors should be carefully considered.

5. Maintenance - Provision for maintenance is a necessity. A good set of drawings and operation instructions must be provided with the system. Remember that the building's maintenance man or operating engineer may be totally unfamiliar with solar energy systems.

A collection of design and installation guidelines is provided below.

1. The Solar Structure

DO provide unobstructed southern exposure.
DO insulate the structure very well and seal off air leaks, remember that conservation is the cheapest "source" of energy.

2. The Collector Mounting

DO mount the collector lower edge well above the average snow depth (roof mounting is ideal), to allow snow run-off.
DO retain a waterproof roof under the collector.
DO use 2 to 5 cm stand offs under the collector to prevent rot under collector.
DO use temporary scaffolding or fork lift to install the collectors.
DO mount the collector with the proper direction running vertical.

3. Collector Piping

DON'T use plastic or PVC piping without careful consideration. Many types only last 5 years. Swimming pool heating systems are an exception.
DON'T use rubber hoses with screw-type hose clamps.
DO use an expansion tank on the closed loop (collector loop).
DO use a sight glass on the expansion tank.
DO make the collector drainable and fillable.
DO use expansion bellows or "U"-tubes for thermal expansion.
DO use highest grade hydraulic hose or equivalent if flexible hoses are necessary.
DO install automatic or manual air bleed valves at the top of the tank.
DON'T plumb in inverted "U" traps which will trap air in the lines.
DO provide means for monitoring the fluid pH and freezing point.

4. System Start-up and Check-out

Do leak-check with compressed air to 207 kPa and soap spray on each solder joint.
DO mount (at least temporarily) a pressure gauge 0 - 207 kPa or higher.
DO fill the system from the bottom up with an auxiliary pump.

5. Storage System

DON'T oversize storage.
DO provide separate footing for storage if necessary.
DO use foam storage insulation if the storage tank is to be buried, i.e. make the insulation waterproof. FRP tanks are O.K. but generally more expensive.

6. The Back-Up Heating Plant

DON'T put the solar heat coil downstream of the furnace.
DO put the fan coil in front of the furnace blower.
DON'T overheat the blower motor.

7.3 CORROSION

The five most common types of corrosion and their methods of prevention and control are discussed below.

1. Galvanic Corrosion

This type of corrosion can result when two dissimilar metals come into contact in the presence of an electrolyte, that is, a suitable liquid. The corrosion on the less resistant (or less noble) metal is increased while the rate of attack on the more resistant metal is decreased. The less noble metal is known as the anode and the more noble metal the cathode. Metals can be listed in terms of their resistance to corrosion, as in the electromotive force or galvanic series. A galvanic series of some commercial metals and alloys is shown in Figure 7.3.1.

Galvanic Series of Some Commerical Metals and Alloys in Seawater.

(Anodic or active end)
Magnesium and magnesium alloys
Zinc
Commercially pure aluminum (1100)
Cadmium
2024 aluminum (4.5Cu, 1.5Mg, 0.6Mn)
Steel or iron
Cast iron
Chromium steel (active) 13% Cr
Ni-Resist (high-Ni cast iron)
18-8 stainless steel (active)
18-8-Mo stainless steel (active)
Lead-tin solders
Lead
Tin
Nickel (active)
Inconel (active: 80Ni, 13Cr, 7Fe)
Hastelloy B (60Ni, 30Mo, 6Fe, 1Mn)
Chlorimet 2 (66Ni, 32Mo, 1Fe)
Brasses (Cu-Zn)
Copper
Bronzes (Cu-Sn)
Cupronickels (60-90Cu, 4-10Ni)
Monel (70Ni, 30Cu)
Silver solder
Nickel (passive)
Inconel (passive)
Chromium steel (passive)
18-8 stainless steel (passive)
18-8 Mo stainless steel (passive)
Hastelloy C (62Ni, 17Cr, 15Mo)
Chlorimet 3 (62Ni, 18Cr, 18Mo)
Silver
Titanium
Graphite
Gold
Platinum
(Cathodic or noble end)

FIGURE 7.3.1

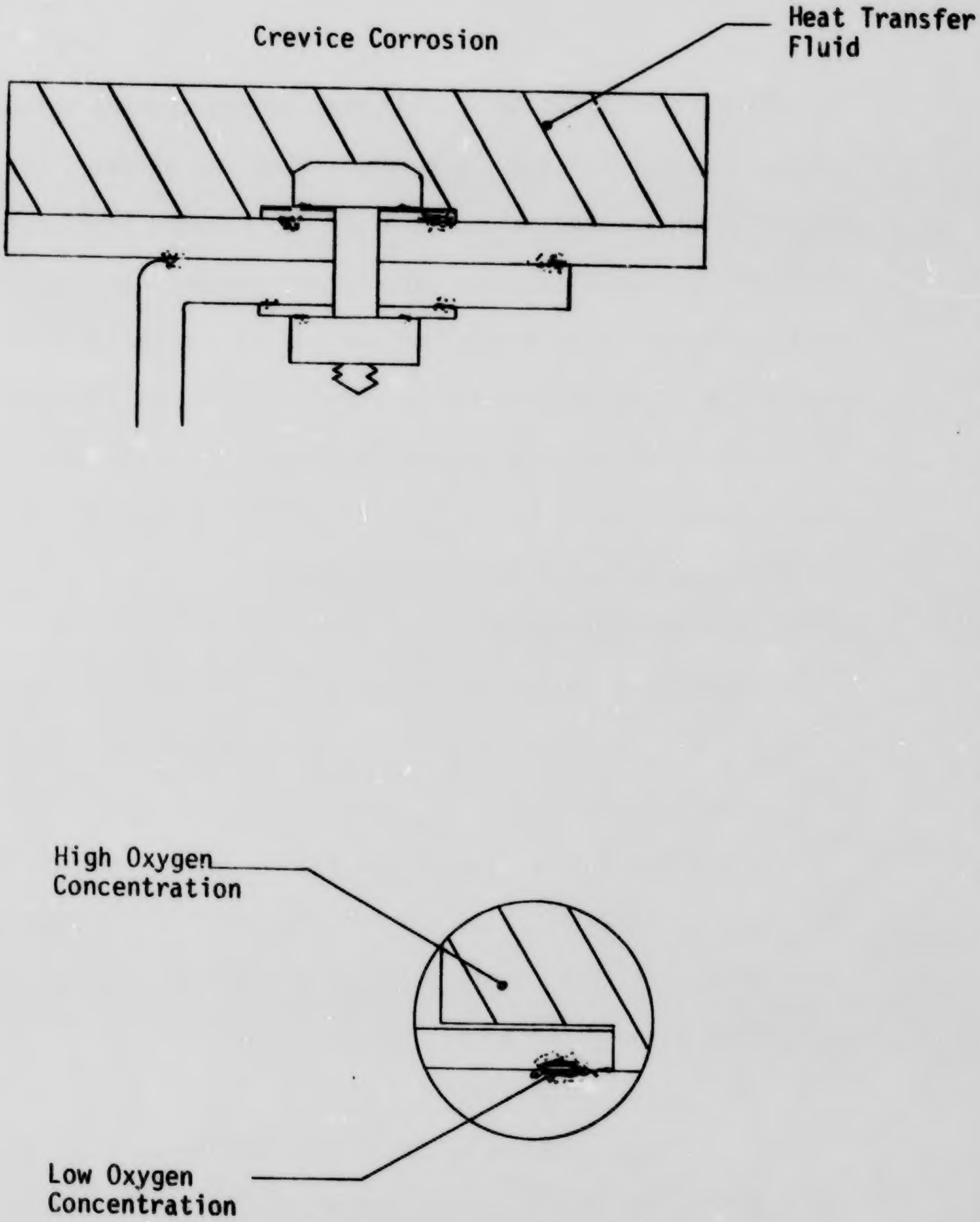
Crevice Corrosion
Heat Transfer Fluid
High Oxygen Concentration
Low Oxygen Concentration

FIGURE 7.3.2

Methods of Prevention - The most common control method is to use metals that are located close together in the galvanic series. Dissimilar metals (such as steel and cast iron) can be used in direct contact if they are located within the same set of brackets in Figure 7.3.1. Another control method is to avoid combinations of dissimilar metals where the area of the less noble metal is small compared to the more noble metal. It is also possible, of course, to electrically insulate dissimilar metals. A protective coating may be applied to the more noble end or chemical inhibitors may be added to the electrolyte.

2. Uniform Attack

This is a general corrosion which proceeds over the entire exposed surface. An example of this would be the simple oxidation of a metal which occurs in air.

To prevent this type of corrosion, durable materials should be selected, suitable inhibitors should be added to any liquids which are in contact with susceptible surfaces and protective coatings should be applied to exposed surfaces.

3. Crevice Corrosion

This is caused by the establishment of local corrosion cells. For example, the difference in oxygen concentration across the washer shown in Figure 7.3.2 can make the surface in contact with the low oxygen solution anodic and the other surface cathodic.

To prevent this corrosion, avoid rough spots where moisture, dirt and other contaminants can collect. Rough spots or rapid transitions in pipes should be avoided. The system should be designed to be completely drainable, thus avoiding pockets of stagnant fluid. Oxygen should be excluded from the heat transfer fluid - for example, through the use of air separators. Non-absorbent gaskets should also be used.

4. Pitting Corrosion

This is an extremely destructive form of localized corrosion which results in deep holes in the metal. Something will initiate a small hole in the material, and then corrosion will proceed rapidly due to the formation of a local concentration cell.

The use of chemical inhibitors (which form films on the metal) or complexing agents (which tie up metal ions in solution) may be used to prevent this type of corrosion. A getter column upstream from the metal which is to be protected can be used to act as a sacrificial anode. Another method is to use deionized water to remove chloride ions which can initiate the pitting. A good strainer should also be used.

5. Erosion Corrosion

This corrosion is a combination of chemical attack and destructive wear. It occurs where the fluid is travelling at a high velocity.

It can be prevented by modifying the shape and geometry of the system such that turbulence is reduced. The fluid velocity should also be limited. The National Research Council of Canada recommends a maximum velocity of 1.5 m/s for water or water solutions in copper tubing and 3.0 m/s for steel tubing. A good strainer should also be included to remove abrasives which can wear down the metal surface.

NOMENCLATURE

A	area, auxiliary, altitude
A_c	collector area
A_a	aperture area
A_r	receiver area
C	cost, concentration ratio, capacitance rate
C_p	specific heat
C_1, C_2	Planck's first and second radiation constants
C_b	bond conductance
D	diameter (defined locally), downpayment fraction
E	energy, equation of time
e	emissive power, base of natural logarithm
F	fin efficiency factor, control function (defined locally)
F'	collector efficiency factor
F''	collector flow factor
F_R	collector heat removal factor
F'_R	collector-heat exchanger factor
	annual fraction by solar energy
f_s	monthly fraction by solar energy, focal length, fraction of days with various K_T
g	gravitational constant
G_{sc}	solar constant, W/m
G	irradiance, W/m
H	daily irradiation, J/m
h	heat transfer coefficient, Planck's constant
I	hourly irradiation, J/m ; can also be considered as an hourly average rate
i	interest rate
K	extinction coefficient

$K_{\tau\alpha}$	incidence angle modifier
k	thermal conductivity, Boltzmann's constant
$\overline{K}_T$	monthly average daily clearness index
L	longitude, length, distance, loss, load
l	length
$\dot{m}$	flow rate
n	day of the year, index of refraction, hours of bright sunshine in a day
N	number of covers, day length, term of mortgage or economic analysis
P_{s_1}	dimensionless collector loss ratio
P_s	modified dimensionless loss ratio
P_L	dimensionless absorbed energy ratio
PWF	present worth factor
Q	energy per unit time, energy
q	energy per unit time per unit area or length
r	radius, ratio
R	ratio of total radiation on tilted plane to that in plane of measurement (usually horizontal), heat transfer resistance
$\overline{R}$	monthly average R
R_b	ratio of beam radiation on tilted plane to that on plane of measurement (usually horizontal)
S	absorbed solar energy per unit area
T	temperature
U	heat transfer coefficient
V	specific volume
W	distance between tubes
x	inflation rate
y	effective interest rate (i-x)

Greek

α	absorptance, thermal diffusivity
γ	surface azimuth angle, bond thickness, intercept factor
γ_s	solar azimuth angle
δ	declination, thickness (defined locally), dispersion
ε	emittance, effectiveness
η	efficiency (defined locally)
θ	angle (defined locally), angle between surface normal and incident radiation
θ_c	acceptance half-angle of CPC
λ	wavelength
μ	absolute viscosity, cosine of polar angle
ν	kinematic viscosity, frequency
ρ	reflectance, ground reflectance, density
σ	Stefan-Boltzmann constant
τ	transmittance, time
ϕ	latitude, angle (defined locally), utilizability
ω	hour angle, solid angle

Subscripts

a	air, ambient, absorbed, aperture, annual
b	blackbody, beam, back, bond, bed
c	collector, critical, cover, corrected
d	diffuse, day
e	effective, equivalent, edge
f	fin, fluid, fuel
i	incident, inlet
L	loss
n	normal, noon

p	plate
r	radiation, reflected, receiver
T	tilted, on a tilted plane
u	useful
z	zenith
λ	wavelength

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Solar Programs Office

SOLAR DESIGN COURSE MANUAL

Part II: ADVANCED

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CHAPTER 1

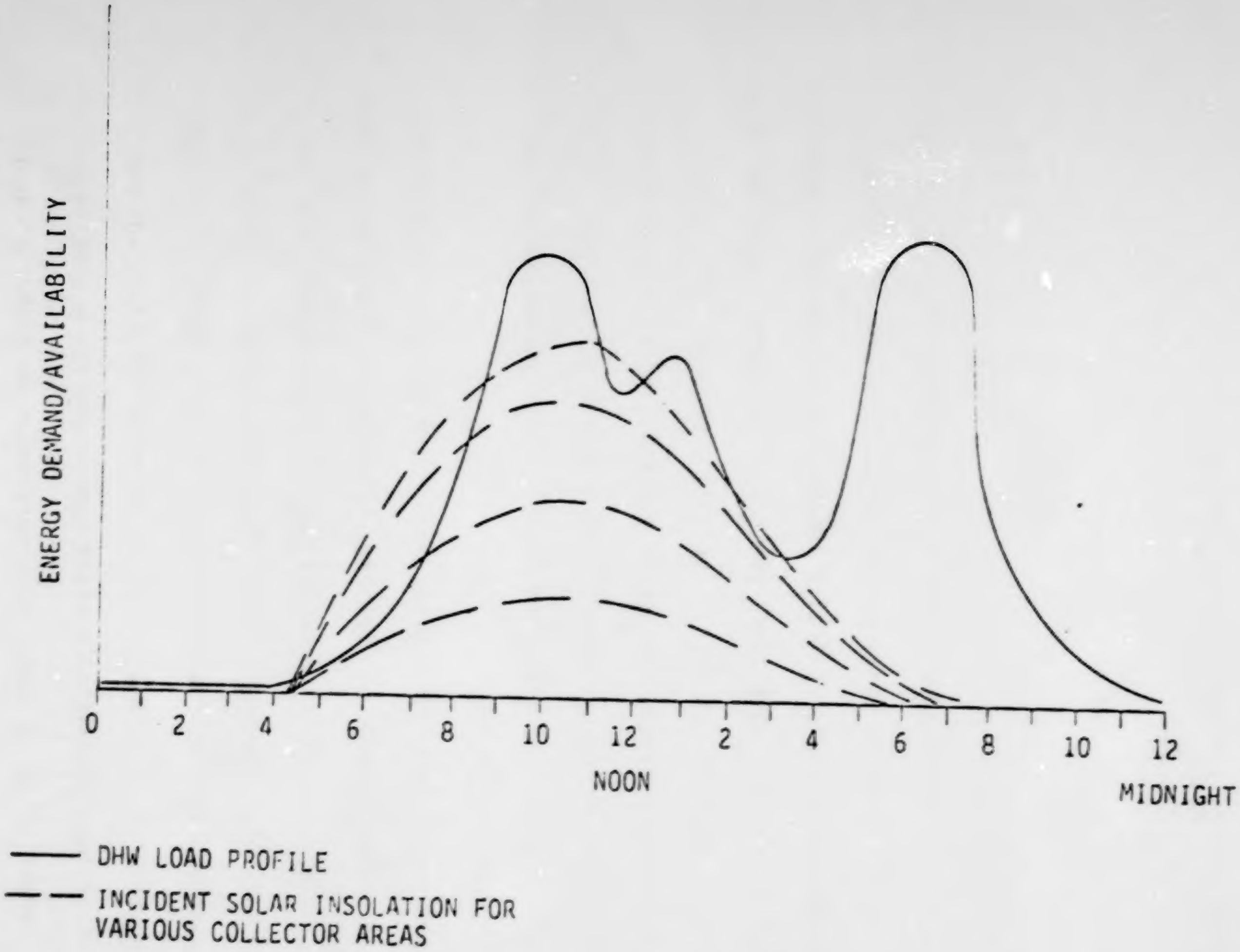
EVALUATING OPPORTUNITIES FOR SOLAR ENERGY SYSTEMS

1.1 OPPORTUNITY IDENTIFICATION

1.1.1 Load Evaluation

When evaluating the opportunities to apply solar energy to meet part of the energy load in a facility, one should look for an application which has a large, continuous, centrally located load which can utilize low temperature energy. At the same time, the impact of energy conservation on the load should be assessed. Energy conservation measures must be performed first to avoid oversizing the solar system. The high fixed costs associated with solar energy systems detract from the economics of small systems. Large loads can justify large solar energy systems, which can benefit from economies of scale.

If there is a continuous demand for solar energy, the problems associated with energy storage are reduced. Thermal storage for more than one day of energy use cannot be economically justified in most applications. A load which is matched to solar radiation availability both on a daily and a seasonal basis will exhibit better economic performance than poorly matched systems. Figure 1.1.1 plots incident radiation against a domestic hot water load profile on a clear summer day for a particular application. Larger collector areas are represented by higher incident radiation curves. At those times when incident radiation exceeds the load,



COINCIDENCE OF DAILY SOLAR
AVAILABILITY & DHW LOAD

FIGURE 1.1.1

the difference between the supply and demand curves is available for thermal storage. As one can see, if the designer wishes to meet only a small fraction of the DHW load, as represented by the lower incident radiation curves, little or no storage is required as the energy available never or only rarely exceeds the energy demand. A larger collector array, meeting a larger fraction of the DHW load will, on occasion, receive more energy than can be simultaneously used. The designer must then decide whether the value of energy outside the use curve justifies thermal storage. If not, he must decide whether the increase in system size is justified when compared to the useful energy gained. When thermal storage is used, the FCHART computer program can assist the designer in selecting the optimum collector storage combination. If storage is not used, then the designer must use other tools, such as the WATSUN Interactive program to optimize collector area.

A load may be evaluated by measuring the actual energy use, or it may be estimated using standard engineering procedures, such as those outlined in ASHRAE. Measured data may come in the form of utility bills, or may be obtained by installing specific instrumentation on the facility meeting the load. The load profile often used for DHW load evaluation is the FCHART profile, based on the TRNSYS computer program and shown in Figure 1.1.2., which is the RAND LOAD PROFILE For RESIDENTIAL USE.*

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* Rand Load Profile, ref. Rand Corporation, J.J. Munch, Residential Water, Heating; Fuel Conservation - Economic & Public Policy - R-1498 NSF, May 1974.

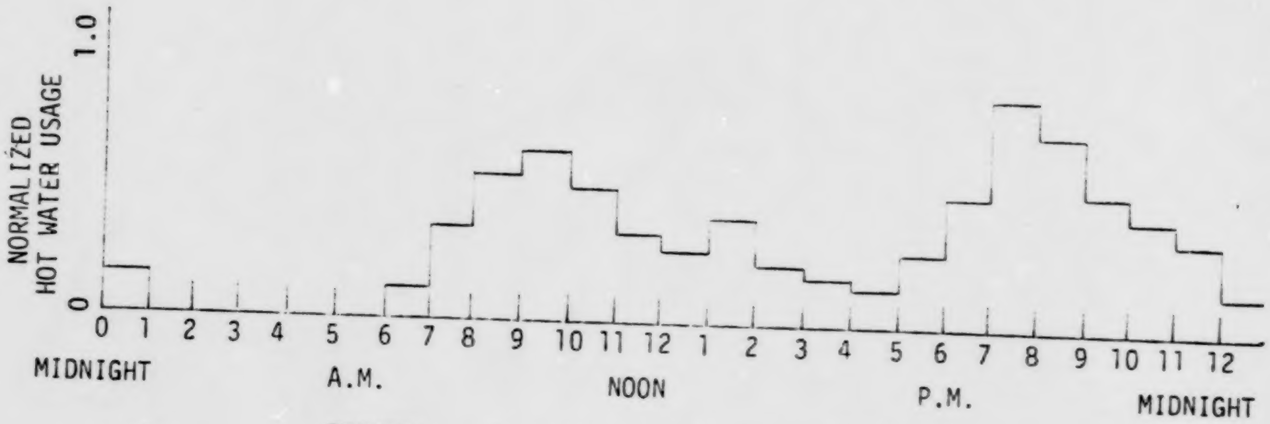

FIGURE 1.1.2 Daily Mass Flowrate Profile

Space heating loads can be evaluated by calculating the heat loss coefficient for the structure and using this coefficient in conjunction with degree day data for the given location or by assuming a furnace efficiency and comparing energy use in the building to historic data to derive a heat loss coefficient. The heat loss coefficient can be used directly in the FCHART 4.0 computer program when predicting the performance of various solar energy system options.

A plot of space heating load versus solar availability would show that peak space heating loads occur at night in small structures while solar radiation occurs during the day. If solar energy is to contribute to the peak heating loads in small structures, thermal storage is required to buffer supply and demand. During spring and fall weather, when heating loads may only occur at night, thermal storage is also required. In large buildings, the time of peak heating loads cannot be generally stated, as building operating procedures and use will determine time of peak space heating loads.

To reduce the cost and complexity of a solar energy system, it is desirable to meet a load at a single, central location. This is generally the case with domestic or process water heating facilities, but space heating applications and ventilation air makeup are frequently located at numerous locations throughout a building, particularly in commercial and industrial installations. The application of solar energy in such instances involves duplication of certain components at each location, which increases overall cost. Such applications may not allow solar energy that cannot be utilized by one system to be utilized by another.

The quality of energy required for a particular application will have a major impact on the economics of the application. Although solar energy systems can provide high temperature energy, there are severe penalties associated with collecting higher temperature energy both in terms of system efficiency and equipment costs. For this reason, applications using low temperature energy are much more favourable than high temperature applications. For example, a space heating system would typically require the solar panels to attain temperatures in excess of 25°C before useful energy is available for contribution to the space heating load. A collector with a performance curve as shown in Figure 1.1.3, a T_i of 22°C (return air from the space being heated) and an outdoor air temperature T_a of -15°C requires an incident insulation rate of 370 W/m² before any useful energy can be extracted from the system. At an incident radiation level of 570 W/m², collector efficiency reaches 30% or could have a useful energy output of 171 W/m², assuming other system losses are negligible. Peak normal radiation levels in Winnipeg and Ottawa are around 710 W/m² in December and 1070 W/m² in January. At the given T_a and T_i, the respective operating efficiencies at these peak radiation levels would be 40% and 54%. However, as collectors are rarely perpendicular to incident radiation and cloud cover frequently reduces radiation levels, it can be seen that a system producing high grade energy will have a low useful energy contribution during those months when heating loads are highest.

If the same collector system were used to preheat ventilation air, its operating efficiency would be much higher, actually equal to the y-axis intercept of the collector performance curve. This occurs because

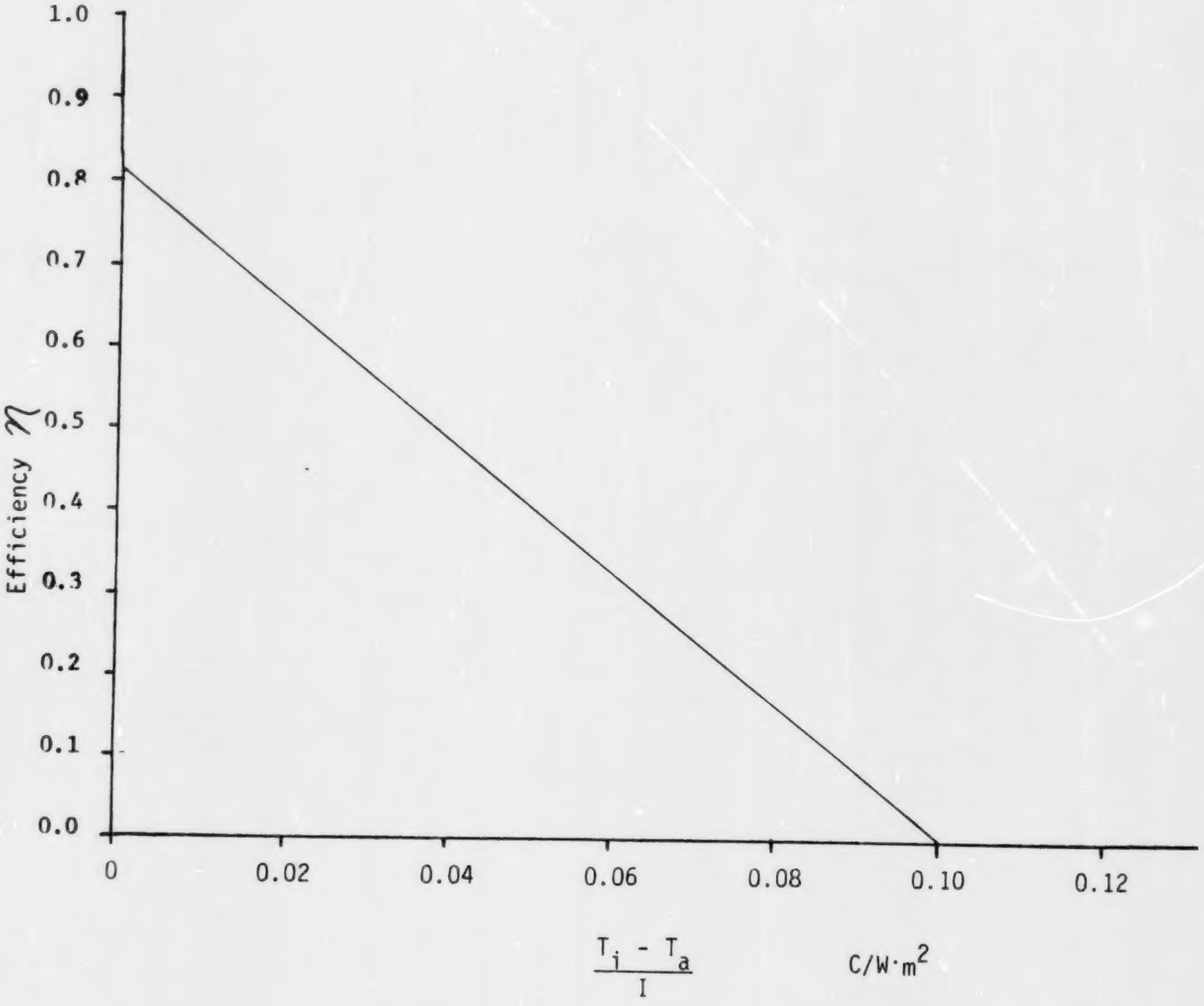

FIGURE 1.1.3

COLLECTOR PERFORMANCE CURVE

the inlet air is outside air, therefore $T_i = T_a$. $\frac{Ti - T_a}{I}$ becomes zero and the collector efficiency becomes the $\tau\alpha$ product. For the preheat application, makeup air would be continuously drawn through the collector whenever there was a preheating requirement, whether or not solar energy is available. When solar energy was incident upon the collector, over 80% of the incident energy would be picked up by the ventilation air passing through the collector. There is no threshold level of incident radiation that must be overcome before the system can contribute useful heat, and the operating efficiency of the preheat system will exceed that of a space heating system under all practical conditions.

The combined effect of lower threshold energy levels and higher operating efficiencies make low temperature applications of solar energy much more favourable than higher grade applications. This is a key factor in identifying solar applications with favourable economic returns. Also using a single solar energy system to meet different levels on a seasonal basis may result in a high annual system performance. For example, air preheating is an ideal winter application for solar energy. Water preheating or swimming pool heating are ideal summer applications, both utilizing low grade energy, relative to ambient air conditions. Occasionally there may be an opportunity to install a single solar energy system to service two loads on a seasonal basis at a cost just slightly greater than servicing either of the loads singly. The result can be a solar energy system with a higher energy performance and cost effectiveness than would otherwise be achieved.

1.1.2 Component Location

The location and orientation of the solar collector array is generally not so critical as one might first imagine. This is important because most systems are designed for existing facilities, and in order to keep costs in line, the solar energy system must frequently utilize building features that are not ideal for solar energy collection. This applies particularly to choosing a location for the solar collector array. In general, it is desirable for a solar collector array to face due south, tilted at an angle equal to the location latitude for a year round application such as domestic water heating, latitude plus 15° for space heating application, and latitude less 15° for summer applications. However, variations in slope and azimuth angles of ±15° from ideals will generally only have a minor impact on system performance, whereas they may result in major reductions in system costs. In fact, variations of collector azimuth of 45° and in some applications 60° from south may only result in reductions of 5% to 10% of annual system performance. This is particularly true of systems designed for summer operation only.

Variations in slope and azimuth may have a positive or negative impact on actual system performance. If morning haze or fog is common, facing the collector array west of south will improve performance. If afternoon smog is typical, an east of south orientation is called for. A flat slope for the collector array will increase the summer solar fraction at the expense of the winter fraction. This may cause overheating in the summer months if summer loads are small.

Shading can significantly reduce the performance of a solar energy system. Aids such as a sunpath diagram (Figure 1.1.4) can be used to show the part of the year and the time of day when an object will obscure direct sunlight on a collector array.

Generally the location of the collector array relative to the other system components is more important than collector orientation. Locating the solar collector array close to the energy user minimizes piping and ducting costs. As ducting and piping constitute a significant fraction of most solar energy projects (10-40%) and are a source of heat loss, their minimization is important. In some instances, it may be possible to use existing mechanical components in the solar energy system. A fan in an existing ventilation system, an existing water line or a second domestic hot water tape can replace necessary solar energy components without adversely affecting the operation of the existing mechanical system.

1.2 OPPORTUNITY OPTIMIZATION

Opportunity optimization involves varying those parameters of a solar energy system which can be manipulated, (eg. collector slope, storage volume, collector type) to find the combination of parameters which will result in the most favourable energy costs over the life of the system. Collector area, for example, is increased to find the point at which the reduction in fixed costs per unit of collector area is less significant than the marginal energy gain attributed to the incremental area. Optimizing collector slope and azimuth can be compared to utilizing existing

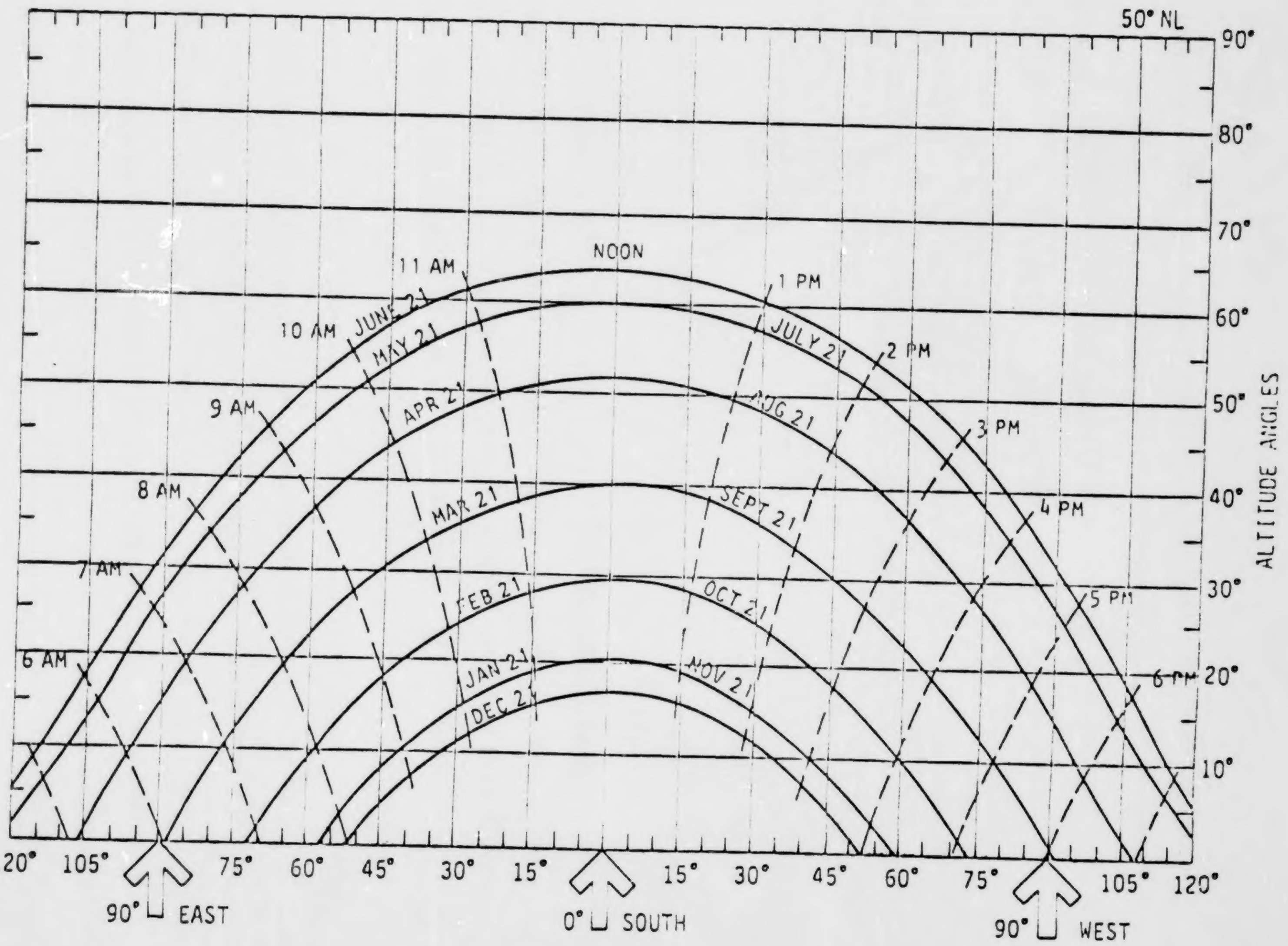

FIGURE 1.1.4

SUN PATH DIAGRAM

building features. The reduced costs associated with collector support structure and mounting may justify the reductions in performance.

The best applications for solar energy are those with a large continuous demand for low grade thermal energy. Low mean collector fluid inlet temperatures result in low heat losses and improved energy collection efficiencies. Low collector inlet temperatures can be maintained if the load being met requires only a small temperature rise, such as the water in an outdoor swimming pool, or if the system is designed to meet a small fraction of a large load, such as pre-heating fresh air makeup in a building ventilation system.

High energy demand allows a system to be sized large enough that fixed and semi-fixed costs can be spread over large collector areas and larger energy contributions, while the fraction of the load met remains low. A low solar fraction combined with a high but fluctuating energy demand may result in a system which utilizes more energy as it is available.

In optimizing system parameters, one should be aware that high quality equipment, with its higher cost, does not necessarily lead to more cost or energy effective solar energy systems. One example of a low cost solar collector possibly out-performing high cost components is an outdoor swimming pool system. The pool temperature is often below the ambient air temperature, due to evaporative cooling of the pool, and to night losses. This means that frequently the solar collector with pool water passing through it will be operating at temperatures below the surrounding air and so rather than suffering heat losses will actually experience heat gains. The more costly, double glazed, well insulated collector will minimize these heat gains, to the detriment of energy gain. A low cost, uninsulated unglazed collector, will experience significant energy gains from the surrounding air. This is shown in Figure 1.3.1. when T_i exceeds T_a, the collector operate to the left of the vertical axis, and efficiencies actually exceed the ideal collector efficiency. Anytime the $\frac{Ti - Ta}{I}$ variable is to the left of the intersection of the two performance curves, the low cost collector would out-perform the higher cost unit. This occurs for significant periods of time during pool season, thus low cost collectors are favoured for outdoor swimming pool installations.

Among the solar applications that may be considered as amongst the best applications for solar energy are: outdoor swimming pool heating, ventilation air preheating (especially in industrial, laboratory or health care type situations where very large amounts of fresh ventilation air are required to replace contaminated exhaust air), and process water heating (in situations where water consumption is high and the make-up water temperature is low).

1.3 EFFECT OF ENERGY CONSERVATION

When evaluating a solar energy application the designer should also evaluate energy conservation options. In general, energy conservation options are economically more favourable than solar options, and so should be considered before solar energy is evaluated for a particular load.

The implementation of an effective energy conservation measure will reduce the load to be met by other means. The effect of other conservation

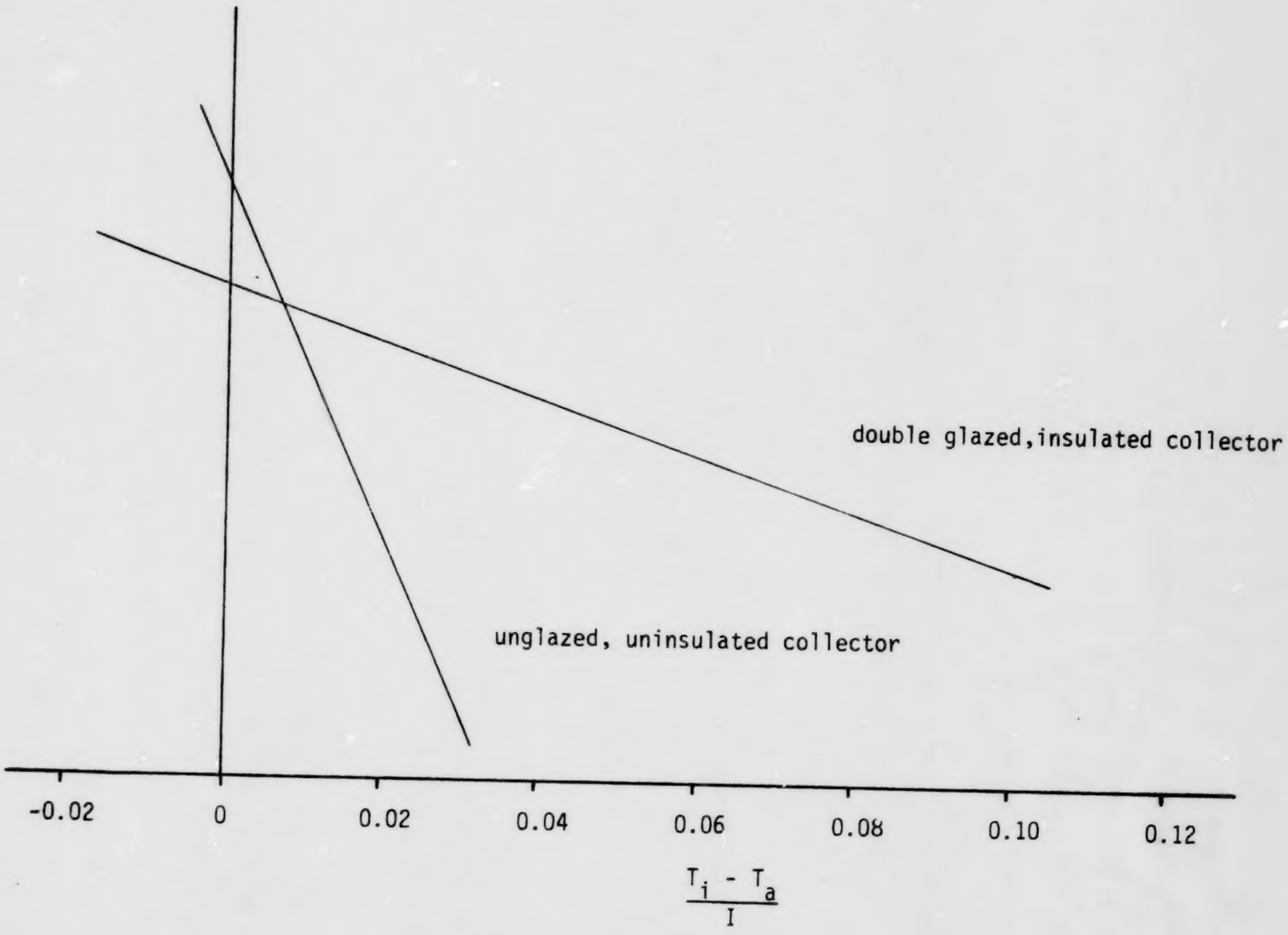

FIGURE 1.3.1

COLLECTOR PERFORMANCE CURVES

measures applied to the same load will therefore be reduced. As the load available to be met by solar energy decreases the energy contribution of the solar energy system will be reduced, thus adversely affecting the economics of the solar energy system.

For example, a DHW system operating with an inlet water temperature of 40°C and a supply temperature of 75°C will have a much higher energy use than the same system with a supply water temperature of 45°C, due to the lower standby losses, and a lower average mixed water temperature. A given solar energy system will contribute less energy to the second system, although it will meet a higher fraction of the DHW energy load. In the first case it may have met 30% of a 100 GJ load while the second case may have met 40% of a 60 GJ load. This reduction in actual energy contribution could mean that the economically optimum system would be reduced somewhat in size, or at least that the economics of the system are less favourable.

1.3.1 PUSH FEASIBILITY REPORT

Before an in-depth evaluation of a solar energy utilization opportunity starts, the facility occupant can complete a Feasibility Report. This report involves selecting a solar energy application and type, quantifying the energy requirements, and then estimating system size, cost and energy contribution using a set of tables and graphs. This report is considered as a preliminary evaluation of the proposed solar energy application. If the proposal is acceptable, the project is funded for a design concept, which is a more comprehensive evaluation.

The Design Concept includes:

- a more complete and detailed description of the load being met by solar energy. Load measurements within a period of time are essential since the solar system greatly depends on the size and profile of the load,
- a proposed system layout, showing the location of major system components,
- a description of the building and any modifications necessary to accommodate the solar energy system,
- a review of energy conservation measures which could be implemented in the facility.

Generally funds are not sufficient for an in-depth energy audit, but the following items should be evaluated:

- resetting DHW temperatures, use of flow restrictors in high use washroom fixtures, and time clocking pumps and exhaust fans where possible.
- a review of assumption and design criteria used in establishing operating procedures.
- a cost estimate, including design, construction supervision, capital costs, expected annual maintenance cost, system energy consumption and any incidental expenditures, such as roof repairs or other building modifications.

The design phase results in a set of structural, mechanical and electrical drawings, specifications and detailed cost and system start up and operational data. Acceptable cost figures based on todays (1982) state of the art, range from 160 $/GJ.a to 350 $/GJ.a.

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CHAPTER 2

2.1 SYSTEM TYPES AND OPERATION

2.1.1 Air Systems

Air collectors have two major applications, space heating and ventilation air preheating. Other applications such as grain drying and DHW heating have also been tried with only modest degrees of success.

In air collectors systems the thermal performance is a function of the collector area. As the collector area increases, the solar fraction will increase as shown in Figure 2.1.1. However, when the utilization per collector unit area is plotted versus the solar fraction a curve such as Figure 2.1.2 results. The curve shows a tendancy towards diminishing returns, as the solar fraction increases the utilization per unit area decreases. It is then desirable to have a low solar fraction in order to achieve a high utilization per unit area so that the thermal performance is maximized.

The solar gain per unit area will vary with the air collector systems application. The highest solar gain per unit area is experienced by the preheating application because the collectors operate at the Y-intercept of Figure 2.1.3. The Y-intercept corresponds to the temperature into the collector being the ambient air temperature. For the space heating application that is not the case and so the solar gain per unit is lower.

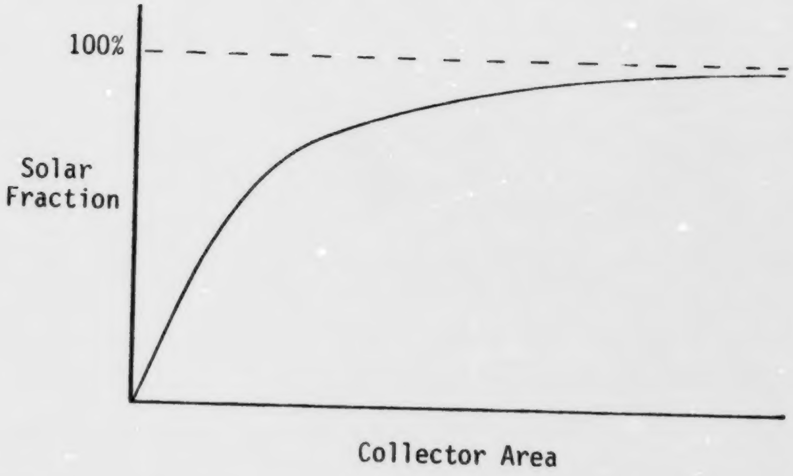


FIGURE 2.1.1 SOLAR FRACTION VS. COLLECTOR AREA

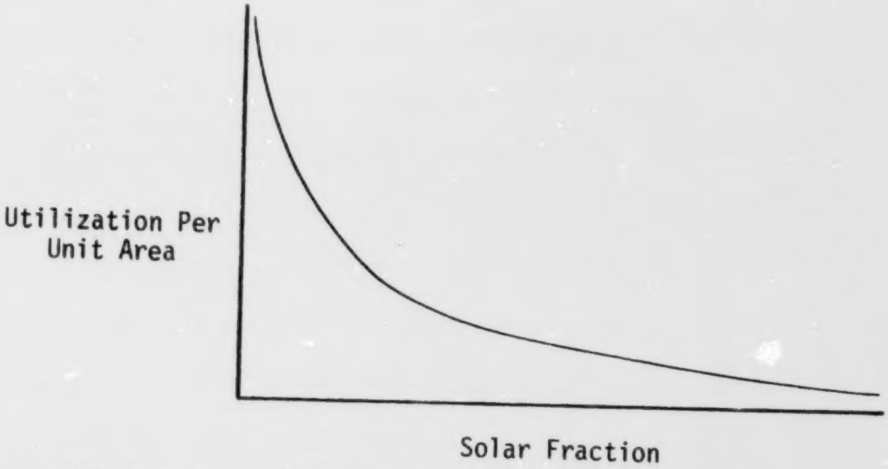
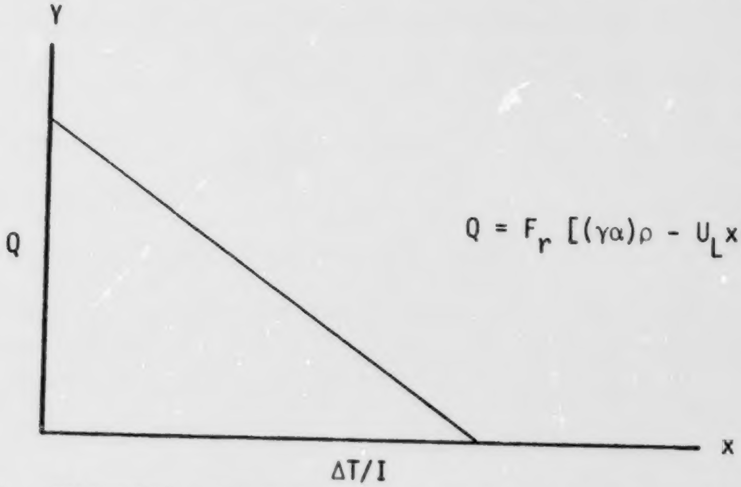


FIGURE 2.1.2 COLLECTOR UTILIZATION PER UNIT AREA VS. SOLAR FRACTION



$$Q = F_r\ [(\gamma\alpha)\rho - U_L x(\frac{T_{in}-T_{amb}}{I})]$$

FIGURE 2.1.3 COLLECTOR ENERGY GAIN VS. THE INLET TO AMBIENT AIR TEMPERATURE DIFFERENTIAL

The majority of manufacturers recommend a flow rate of 0.01 m^3/s per m^2 of collector area; with one recommendation for the flow rate ranging from 0.01 to 0.03 m^3/s per m^2. The number of collectors that can be safely arranged in series is a function of the pressure drop through the total system. The recommended pressure drop through a collector system should not exceed 0.625 kpa and so the exact number of collectors that can be used is dependent on the manufacturers rated pressure drop.

The pressure drop per collector varies from collector to collector. Currently manufactured collectors have a recommended operating pressure ranging from 0.02 to 0.50 kpa. Collectors operating at the lower pressure make it possible to combine several collectors in series without exceeding the 0.625 kpa pressure drop limit. The collectors having the high operating pressures can not be combined in series without exceeding the pressure drop limit and so a parallel collector array would be required.

The majority of solar systems require more than one solar collector and so several combinations of collectors can be arranged to form a larger collector array. The most straight forward arrangements are series and parallel. At present there are methods for calculating the performance of N collectors in series while using manufacturers data for a single collector.*

In order to maximize the efficiency in a series array of collectors the manufacturers recommended flow rate must be maintained in each collector. However when collectors are placed in series the pressure drop of each collector adds together which effectively reduces the airflow rate. And so if N collectors are placed in series then the pressure drop through the collector array will be approximately equal to N times the pressure drop of one collector. Because the pressure drop increases N times the airflow rate must also increase N times in order to maintain the airflow through each collector. The increase in airflow results in the fan horsepower also having to be increased with the increase in fan horsepower being approximately equal to the cube of the number of collectors in series. This results in a larger fan than would be required for a single collector.

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* Duffie & Beckman - Solar Engineering of Thermal Processes, pp. 268-270.

This is not the case for parallel collector arrays since the pressure drop per collector does not add together as in a series collector array and so a parallel collector array can employ a smaller fan. However tests by Dunkle and Davey (1970) observed that collectors in parallel experienced variable pressure drops, depending on the collectors location in the parallel array. It was observed that the pressure drop from bottom to top of the collector was greater at the ends of the array than at the center portion of the array. The irregular pressure drops indicate a lack of uniform flow in the collectors and so the performance of collectors in parallel suffers. The reduction of flow results in reduced collector efficiency. The selection between series and parallel collector arrays then becomes a choice of running a larger fan for a series array or running the parallel array system longer to overcome the reduced collector efficiency. Dunkle and Davey recommend connecting the collectors in a series parallel or multiple parallel arrangement which results in more uniform flow distribution. Other considerations when choosing between parallel and series collectors arrays include manifolding and the collectors design.

2.1.2 Effect of Storage Size on Thermal Performance

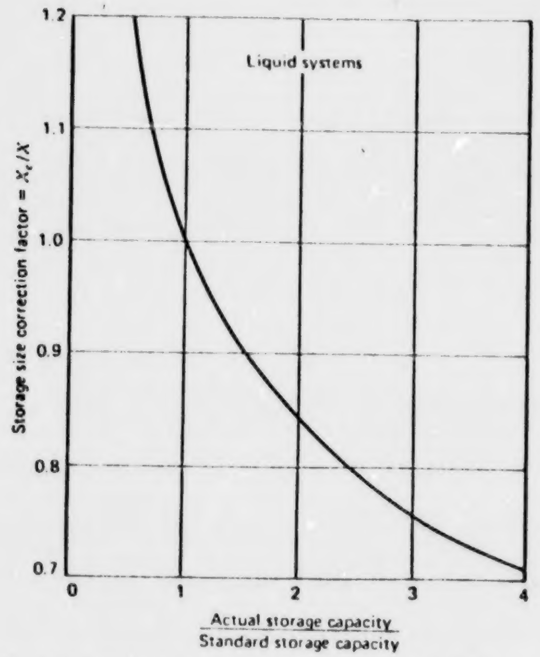
When air collectors are used two forms of thermal storage can be employed; sensible and latent heat storage. The storage media choosen is dependent on whether the heat is stored as sensible or latent heat and example being a pebble bed heat exchanger for sensible heat storage. The major characteristics of a thermal energy storage system are (a) its capacity, per unit volume or weight; (b) the temperature range over which it operates; (c) the means of addition or removal of heat and the temperature differences associated therewith; (d) temperature stratification in the storage unit; (e) the power requirements for the addition or removal of heat; (f) the containment associated with the storage system; (g) the means of controlling thermal losses from the storage system; (h) its cost. These characteristics hold true for all the energy storage methods employed in solar energy systems.

In a packed bed storage a bed of loosely packed particulate material is used to store energy. A variety of solids can be used with rock being the most widely used material. A packed bed storage using rocks will have a high heat transfer coefficient between the air and the solids, which promotes thermal stratification; other advantages of a rock bed storage are the low cost of storage materials and the container; the conductivity of the bed is low when there is no air flow; and the pressure drop through the bed is low. With these advantages the use of a rock bed storage system is the best suited for an air collector system.

The performance of air systems is less sensitive to storage capacity than that of liquid systems. The storage volume required by an air collector system is dependent on the systems application. Air systems can operate in the collector-load mode, in which the storage unit is bypassed. Pebble bed storage is commonly employed with air collector systems because they are highly stratified and additional capacity is effectively added to the cold end of the bed because it is seldom heated and cooled to the same extent as the hot end.

The storage volume employed by the FCHART computer program for air collector systems is 0.25 cubic meters of pebbles per square meter of collector area, which corresponds to 350 kJ/m²C for typical void fractions and rock properties. The performance of systems with other storage capacities can be determined by employing a storage size correction factor, X_c/X, as indicated in Figure 2.1.4, which was taken from Solar Engineering Of Thermal Processes by John A. Suffie and William A. Beckman

A basic method for determining the spacing between rows of collectors is that the shade angle should not exceed the angle of the sun on December 21 at 12:00 noon. The shade angle is defined as the angle which is formed when a line from the top and the bottom of the southern most row of collectors intersects the bottom of the next row of collectors. If exterior manifolds are used then the top of the exterior manifold becomes the highest point when calculating the distance between collectors using the shade angle. Although this would be ideal spacing between



Storage size correction factor for liquid systems. Standard storage capacity is 75 liters/m².

FIGURE 2.1.4

collector rows, other factors must be considered such as the building dimensions, design and orientation. The building dimensions limit the spacing between collector rows, especially when several rows of collectors are employed.

2.1.3 Storage: Advantages and Disadvantages of Rock, Liquid and Phase Change

Good performance of solar heating systems during periods of low or no sunlight is critical if a significant benefit is to be realized. The capability to store substantial amounts of thermal energy for long periods of time is essential.

The two major classes of thermal storage media are those which store heat as sensible heat and those which store heat as latent heat of fusion. Sensible heat is the thermal energy contained by a material, with its heat capacity defined as the quantity of heat which is absorbed or released when the temperature of 453 grams of the material changes 0.555°C.

The latent heat of fusion is the energy that is absorbed when a material melts and released when the material solidifies. The advantage of latent heat storage over sensible heat storage is that a greater quantity of energy can be stored in a smaller volume of material at a convenient operating temperature. Latent heat of fusion materials are commonly referred to as phase change materials or PCMs while the sensible heat media include water and rock.

The most common sensible heat storage media are water and rock but numerous others exist such as water/glycol mixtures, scrap iron and aluminum, concrete, brick and paraffin oil. Rock is probably the most commonly used storage medium in air systems. A major disadvantage of rock storage as compared to a water storage medium is that rock has approximately one-half the heat storage capacity of water. In order to store an equivalent amount of thermal energy, twice as much rock must be used as compared to water. However, the need for additional storage space is compensated by the elimination of freezing problems, the danger of boiling or the possibility of flooding living space.

The other drawbacks of using a rock storage medium include packing and channeling of the rock, potential for fungi build up if moisture is present in the containment chamber, and leakage of air from the containment. For maximum heat storage capacity the rocks should be of uniform size and distribution.

Water is the other common sensible heat storage medium. Of the liquid sensible heat storage mediums, water has the highest heat capacity while a 30 percent water/70 percent glycol mixture would have a heat capacity approximately 80 percent that of straight water. When a liquid storage medium is used several problems can occur that are not experienced by solid storage media. The most obvious problems unique to liquid storage would be leaks, water seepage, flooding, freezing and boiling. These problems can be reduced with the proper selection of storage tank materials. An example of a liquid storage tank is a reinforced concrete outer shell with a waterproof liner. The waterproof liner would eliminate water seepage and leaks while the outer concrete shell would allow the tank to be used both above and below ground and also provide high load resistance.

The phase change storage materials (PCMs) can be classified into three general groups: a) salt hydrates, b) organics and c) eutectic mixtures.

PCMs offer an attractive alternative to sensible storage media when there is a problem in providing space for thermal storage medium. PCMs require only a fraction of the space needed to store an equivalent amount of thermal energy; an example being that 3.5 m³ of sodium sulfate decahydrate can store the same amount of energy as does 62.3 m³ of rock or 28.3 m³ of water. Despite this significant reduction in storage space, the various disadvantages of PCMs must also be considered.

The advantages of the salt hydrate group are that they tend to be nontoxic, nonflammable, readily available, and inexpensive. However they do not reliably cycle from liquid to solid and some supercool, that is they fail to solidify when the temperature drops below their melting points, and so do not release the latent heat of fusion at the anticipated temperature.

Organics do not have the precipation of solids or supercooling problem, but they do have a relatively high cost and some may be fire or explosive hazards. The organics have a relatively low latent heat of fusion and so require higher volumes than salt hydrates.

2.1.4 Leakage and Thermosyphoning

In all solar collector systems, leakage of either liquid or air can be a problem. Air collector systems experience air leakage at the dampers, along the ducting and even within the collector. For minimum air leakage the ducts should have taped or sealed joints and be made of fiberglass board insulated sheet metal or flexible insulated duct. A rating of R-4 is recommended as the minimum for duct insulation which corresponds to two and a half centimeters of fiberglass insulation. For ducts, the amount of air leakage is dependent on the length of the ducting, however approximately 5 percent of the air leakage will occur at the inlet and outlet of the ducting.

A second source of air leakage is at dampers. Automatically controlled dampers are employed in air solar systems for controlling the direction of air flow through the rock bed and to the collector and space heating loops. The major problem with dampers in solar air systems is excessive leakage.

The freezing potential for air collector systems with a domestic hot water preheat option is directly affected by the air damper leakage rate. A ten percent leakage in the damper between the heat exchanger and the collector could freeze and rupture the heat exchanger in less than an hour on a cold winter night. In order to minimize leakage, dampers with a leakage rate not exceeding 1/2 percent of the rated flow should be specified.

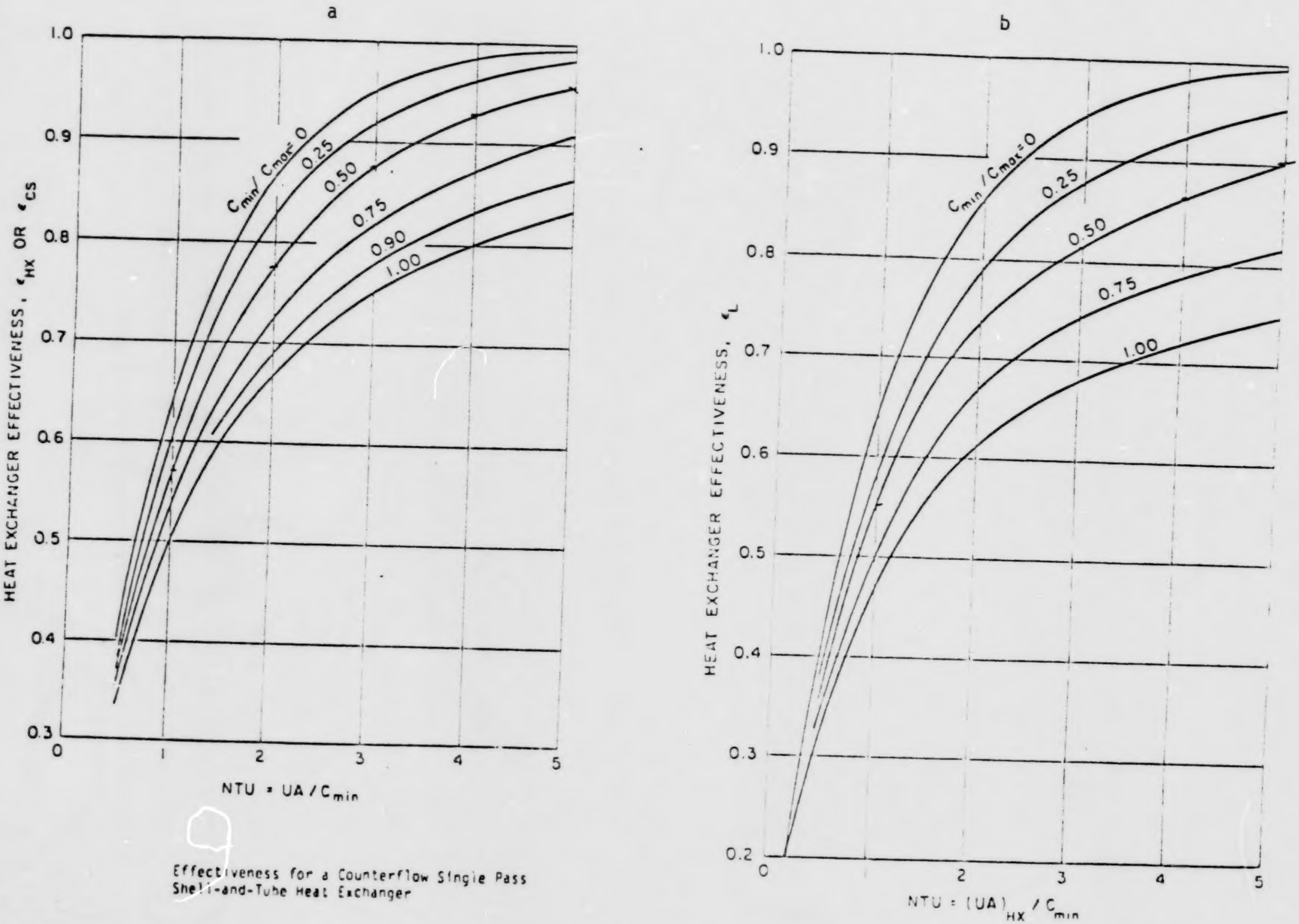
Air leakage on to the heat exchanger can also be the result of thermosyphoning at night. If air is allowed to leak through the collectors at night when the ambient temperature is below the temperature of the conditioned space, then the collectors will dump heat to the colder environment. This will result in significantly reduced thermal performance. Back draft dampers can be used in the ducts to prevent reverse flow and should be of the flexible flat type or shutter type. They should also be mounted to provide a positive seal against reverse air flow. Reverse thermosyphoning can cause damage to a heat exchanger and also reduce thermal performance but these problems can be reduced with properly installed dampers.

A third source of air leakage is from the collectors. Air leakage from the collectors is dependent on the pressure differential between the collector and the ambient air. A negative collector to ambient air pressure differential will result in infiltration to the collector and so the building becomes pressurized. A positive pressure differential results in the building depressurizing and so air is forced out of the collector. Because of this dependence on pressure differential no typical air leakage rate can be assigned to collectors. Since air leakage reduces an air system's performance, it becomes important to check for air leaks before any insulation or covering is applied to the ducts. This can be done when testing the system by using a smoke pencil to check for leaks.

2.1.5 Heat Exchangers

A heat exchanger is used to transfer heat from collector fluid to storage fluid. Perhaps the simplest liquid to liquid heat exchanger is the shell and tube type. For solar systems a counterflow type is appropriate as it minimizes the temperature differences between storage and collector exit fluid temperatures.

A method of sizing heat exchangers which is dependent on the entrance temperatures of the hot and cold fluids, utilizes a factor called heat exchanger effectiveness. Heat exchanger effectiveness is the ratio of



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Effectiveness for a Counterflow Single Pass Shell-and-Tube Heat Exchanger

Effectiveness of Cross-Flow Water-to-Air Heat Exchanger

FIGURE 2.1.5

actual rate of heat transfer to a theoretical maximum rate that would occur if the heat exchanger surface area were very large. The heat exchanger effectiveness can also be expressed in terms of the material used in the heat exchanger and the fluid capacitance rates. The effectivenss for a counterflow single pass shell-and-tube heat exchanger is shown in Figure 2.1.5a. The heat exchanger material and the fluid capacitance rate is used to calculate the number of transfer units, NTU, which when plotted against the ratio of minimum over maximum heat-capacity flow rate, can be used to determine the heat exchanger effectiveness. A similar method can be used to determine the heat exchanger effectiveness of a water-to-air heat exchanger. The effectiveness of a cross-flow water-to-air heat exchanger is shown in Figure 2.1.5b. By comparing the two graphs of Figure 2.1.5 an approximate relationship between the effectiveness of a liquid-to-liquid versus a liquid-to-air heat exchanger can be made. All of the ratios of the minimum over maximum heat capacity flow rate, except for the 0, resulted in a higher heat exchanger effectiveness for the shell-and-tube heat exchanger, for all the number of transfer unit values. From the graphs it appears that the largest difference in heat exchanger effectivness occurred at 5 NTU's and at the 1.0 curve C min/C max, where C is the heat capacity flow rate (i.e. mass flow rate of the fluid multiplied by specific heat of the fluid). Heat exchanger effectiveness is the ratio of actual rate of heat transfer to a theortical maximum rate that would occur if the heat exchanger surface area were very large (infinite). At that point the liquid-to-air cross flow heat exchanger had an approximate heat exchanger effectiveness of 0.74 while the liquid-to-liquid heat exchanger effectivenss was 0.83. From the curves for these two specific types of heat exchangers the liquid-to-liquid heat exchanger consistantly obtains a higher heat exchanger effectiveness. There are however other configurations of heat exchangers but the liquid-to-liquid types should experience higher effectivness than comparable liquid-toair heat exchangers.

The cost disadvantages of custom made equipment as opposed to off-the-shelf equipment can be large. When custom made equipment is used both the design and manufacturing process cost will increase several times. Since custom made equipment implies a one of a kind system, no assumptions common to solar systems can be employed by the design engineer and so additional engineering design time will be required. The assembly of custom made collectors by manufacturers would be several times the cost of the more cost effective mass produced collectors. The same would be true of the solar system components and so the equipment cost could be several times higher than the mass produced off-the-shelf equipment. The higher cost disadvantage of custom made equipment limits its application to very few systems and so custom made equipment should not be considered except for extreme circumstances.

2.2 LIQUID SYSTEMS

2.2.1 Introduction

The principal applications of liquid cooled collectors are for air preheating (or heating) and water preheating (or heating). Preheating applications will result in higher collector operating efficiencies than when solar energy is employed for heating to final conditions, in a given situation.

The simplest system type is the thermosiphon, in which solar collectors are located below the thermal storage. Natural convection circulates the solar fluid from storage through the collectors when the collector temperature is above that of the thermal storage. Thermosiphons require no circulation pumps or controls. The physical position of the components in thermosiphon systems is critical for satisfactory system operation.

Due to physical constraints on the location of components in most liquid systems, pumps are usually used to induce fluid flow through the solar energy collection system rather than relying on free convection. Controls are required to coordinate pump operation with energy availability.

When thermal storage is used it is usually water. Systems in which this water is circulated through the solar collectors are called direct heating systems. In all Canadian locations, some form of freeze protection is required for liquid collectors which operate year round. For direct water heating systems this may be achieved by automatically draining the water from the solar collectors anytime there is a risk of freezing. In space heating applications, an antifreeze solution may be used as a solar fluid and for thermal storage. Indirect or dual fluid systems circulate an antifreeze solution through the solar collectors and use a heat exchanger to transfer the energy collected to water for storage or use.

Although liquid cooled collectors may be used for space heating applications, the high operating temperatures required for most space heating applications degrade system performance. The high cost of interfacing a solar space heating system with the conventional system further detracts from its feasibility. Air systems tend to be similar and less expensive than liquid systems for air preheating applications.

The most common uses for liquid collectors are swimming pool process or domestic water heating and preheating. The advantage to these applications is that the load commonly occurs when solar energy is available and in many instances any energy available in excess of 10°C is useful. Space heating requires 20°C or higher collector fluid temperatures. The technical complexity of systems varies greatly. In general, the cost of energy collected can be reduced by keeping the system simple and designing it to meet a small fraction of a large continuous load.

2.2.2 Effect of Heat Exchangers on Performance

A heat exchanger is required to transfer energy between the two fluids in indirect (dual fluid) systems. If the fluid to be heated is water, then the water is normally used as the energy storage medium. In this case the heat exchanger will be in the collector loop, as shown in Figure 2.1.6.

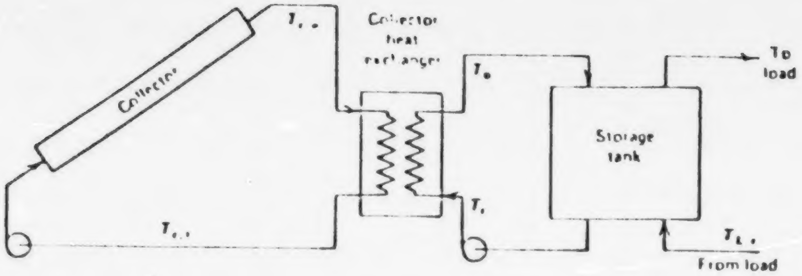
Collector
T_{co}
Collector heat exchanger
T_o
To load
Storage tank
T_i
T_{ci}
T_{L}
From load

FIGURE 2.1.6

HEAT EXCHANGER IN COLLECTOR LOOP

The transfer of heat between the two fluids requires a temperature difference and thus results in a loss of available energy. For a given storage tank temperature, the inlet temperature to the collector will be higher and its efficiency lower. This effect can be accounted for by correcting the value of the collector heat removal factor for the collector (F_R), to a new value given by

$$F_{R(new)} = \frac{F_R}{1 + \left(\frac{A_c F_R U_L}{(\dot{m}C_p)_c}\right)\left(\frac{(\dot{m}C_p)_c}{\varepsilon(\dot{m}C_p)_{min}} - 1\right)}$$

where A_c = collector area

U_L = collector loss coefficient

$\dot{m}$ = mass flow rate

C_p = specific heat

ε = heat exchanger effectiveness

subscript c = for the collector fluid

subscript min = the lower of the values for the two fluids passing through the heat exchanger.

The heat exchanger effectiveness is defined by

$$\varepsilon = \frac{Q}{(\dot{m}Cp)_{min} \times (T_h - T_c)}$$

where Q = the heat transfer rate

T_h = entering temperature of the hot fluid

T_c = entering temperature of cold fluid

Figure 2.1.7 shows the magnitude of this effect. For a typical collector with $(\dot{m}C_p)_c = (\dot{m}C_p)_{min} = 160$ W/°C

$F_R U_L = 4 W/m^2$ °C

and $A = 2m^2$

a heat exchanger effectiveness of $\varepsilon = 0.6$ will result in a decrease in performance of only 3%. This effectiveness is easy to obtain in a counterflow heat exchanger. In a crossflow or concurrent heat exchanger, more surface area will be required, and in a free convective heat exchanger, such as the jacketed tank type, the effectiveness will be much lower and the impact on system performance will be considerable.

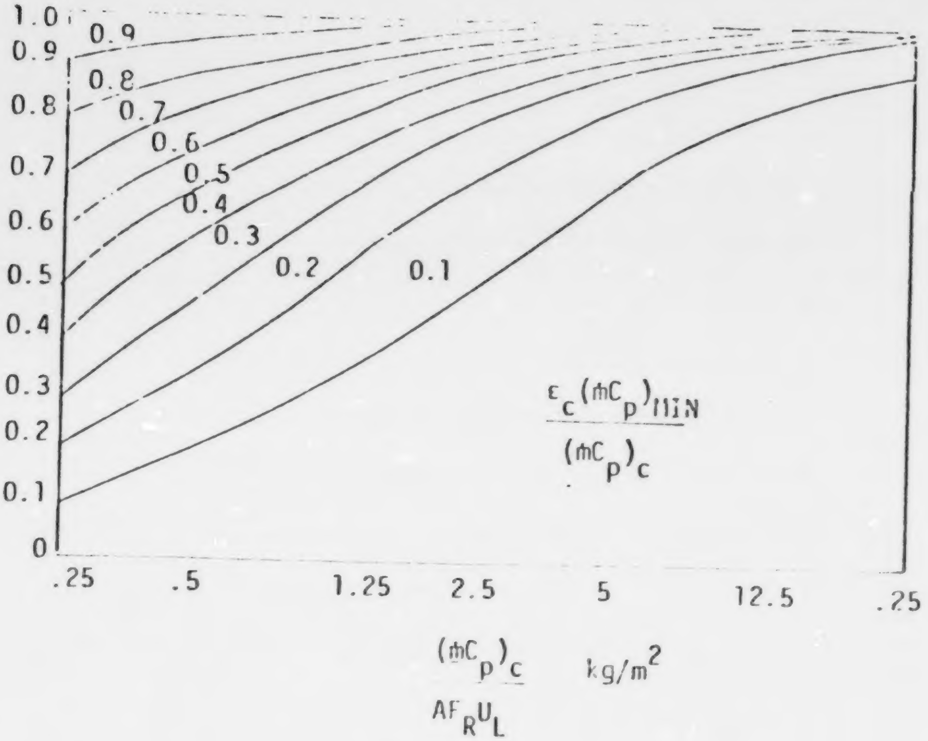

FIGURE 2.1.7 COLLECTOR HEAT EXCHANGER CORRECTION FACTOR

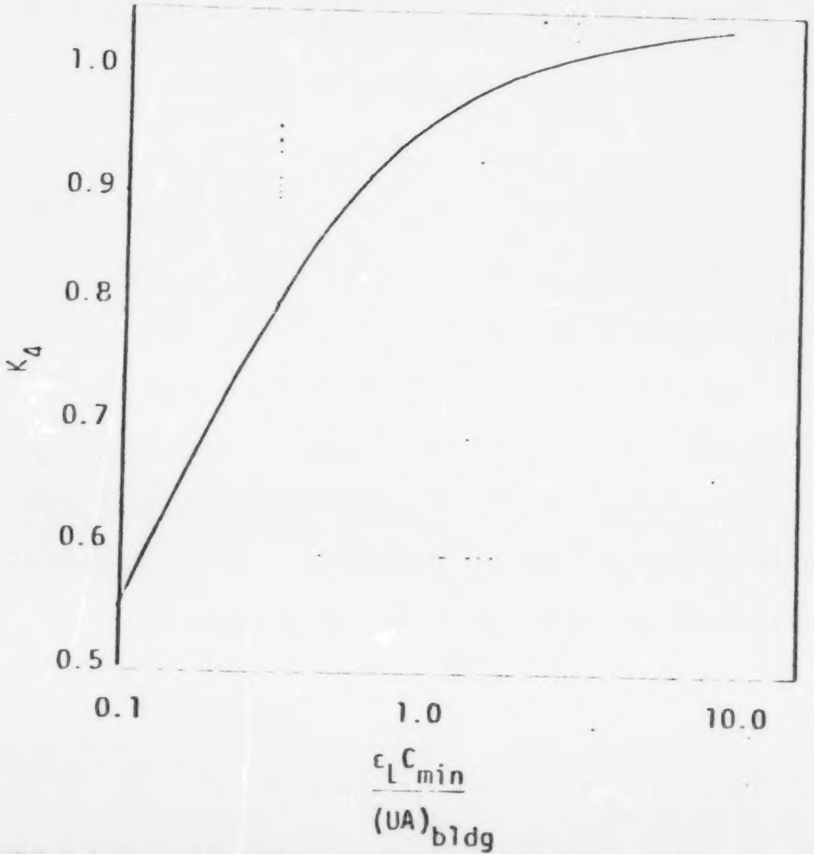

FIGURE 2.1.8 LOAD HEAT EXCHANGER FACTOR, K_4

If a liquid cooled collector is used for space heating, the storage will often be in a liquid and there will be a heat exchanger between the storage and the load. This heat exchanger will also result in a decrease in system performance. A designer who is taking a sophisticated approach to the optimization of such a system will probably use hour by hour simulation to analyse the effect of the performance of this heat exchanger on the systems performance. A simpler approach, taken from FCHART, can provide the designer with an estimate of the impact of the load heat exchanger. The y-intercept, $F_R(\alpha\tau)$ of the collector efficiency curve is multiplied by a correction factor defined by

$$K_4 = .39 + .64 \text{ exp. } \left(\frac{-.139 \, (UA)_{bldg}}{\varepsilon \, (\dot{m}Cp)_{min}} \right)$$

where $(UA)_{bldg.}$ = the thermal conductance of the building and the other variables are defined as above.

Figure 2.1.8 presents this correction factor graphically. This correction is valid in the range $.5 < \dfrac{\varepsilon(mCp)_{min}}{(UA)_{bldg.}} < 10$

2.2.3 Losses From Liquid Cooled Solar Energy Systems

The two main types of solar energy system losses are heat losses from lines, tanks heat exchanger, etc., and parasitic energy consumption. The latter term refers to energy consumed by the system to make it operate, such as pumping power and power to operate control systems.

It is difficult to generalize about the significance of either of these parameters. The amounts vary widely from one design to another. The best way for the design engineer to develop a feel for the significance of these numbers is to calculate them for each system he designs.

A review of several service water heating systems indicates that the parasitic power can vary from about 2% of the useful energy collected for large well designed systems to close to 10% for some smaller systems. Heat losses from the system also vary considerably, with a typical figure for a well designed system being about 10%.

The job of optimising the insulation thickness is a straightforward one which the mechanical design engineer faces repeatedly. In the design of a solar energy system the task is easier than normal because there is no need to speculate on future fuel prices or interest rates, it must be decided whether spending another $1,000 on solar collectors will gain more energy than spending another $1,000 on insulation will save.

As the percentages listed above indicate, parasitic power and heat losses are by no means negligible in a solar energy system, and the careful designer should be sure that he does not allow either of them to become excessive.

2.2.4 Methods of Operating the Solar Energy Storage Tank

The design and operation of the storage system has a major influence on the thermal performance of a solar heating system. The temperature of the liquid leaving the storage system controls that of the collector coolant leaving the heat exchanger and entering the collector and so controls the collector efficiency. Thus the lowest possible temperature at the outlet of the storage tank will result in the best possible system performance.

For a given load and a given solar energy system, the lowest storage tank outlet temperature occurs when the liquid in the tank is kept stratified. This should be a major objective of the system designer. The thermodynamic significance of tank stratification, and ways to achieve it in practice are discussed further in Chapter 3.

One of the problems which designers face is choosing the proper tank size. In a space heating system the issue is a simple one, because the storage tank fluid recirculates, so that all the heat stored above the storage temperature can be used, while none stored below is used. Thus the larger the tank the more usable energy it can store, and the sizing of the tank is an economic optimization problem.

In a service water heating system the problem is different. The designer must decide whether it is better to maintain storage at a temperature higher than the set point and temper the water to the required temperature, or to size the tank large enough that the temperature never approaches the set temperature. There is little published information on this topic, but one study* led to the conclusion that the storage tanks should hold about one day's water requirement.

It is not difficult to see why there should be a thermodynamic optimum size for the storage tank in a solar domestic hot water system.

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* G.K. Yuill and B.C. Bradley, "The Modelling of a Solar Domestic Water Heating System" System Simulation and Economic Analysis for Solar Heating and Cooling Conference, San Diego, 1978.

Energy can be extracted from the tank only by using water. Thus, if 400 L of water at 60°C is required, and the make up water temperature is 10°C, a tank containing 400 L of water at 60°C can supply the entire load. A tank containing 5000 L of water at 50°C contains 10 times as much energy (measured above the inlet water temperature) but it can supply only 80% as much as the smaller tank when the 400 L is drawn out. Thus it is better to use the smaller tank and allow it to exceed the set point temperature of the load on bright summer days, rather than to use a very large tank.

If this approach is taken, the tank temperature will occasionally rise to well above the set temperature of the service hot water system. This creates several problems. One is that the low efficiency of the collectors at the higher inlet temperature may reduce the energy gain to less than the parasitic energy consumption and the line losses in the primary loop. A second is that the high temperature may endanger users of the water. Finally, the higher temperature may damage system components or cause a pressure increase in the system.

The first problem is not a serious one. Most control systems are set to shut off at a slight positive temperature difference, so that parasitic power will not be wasted. Since parasitic power is independent of temperature, the same control setting will be adequate at all temperatures.

One possible operating mode for a solar energy system will run the system even when it is losing energy. Primary loop coolant leaves the heat exchanger at some temperature T_0, and loses heat in the collector supply line. It gains heat in the collectors to rise to a temperature just above T_o. It then loses more heat in the collector return line to return to the heat exchanger at a temperature less than T_c. Thus the primary loop coolant cools the storage tank water instead of heating it in the heat exchanger, but the circulating pumps continue to run because the collector outlet is hotter than the storage tank. This problem can be eliminated by raising the temperature difference required to keep the pumps operating, or by increasing the insulation level on the primary circulation lines. The problem is not increased by using a smaller, hotter tank, since for every tank temperature in any given system there will be a narrow range of insolations which can cause the system to operate in this way, if the turn-off temperature difference is set too low. This narrow range of insolations will occur at least once every day the system operates.

The second problem with the use of small, higher temperature tanks is the risk of scalding. This can occur if the tempering valve is not fast acting enough, for example. When no water is being used and the temperature valve is in the "full hot" position. If a tap is turned on, hot water will move down the line and some will pass through the tempering valve before it has time to react. This could result in injury to the water user.

Scalding by water from domestic hot water systems is a significant cause of injury in Canada. About 300 young children, and a large number of adults, receive medical treatment for third degree burns received in this way every year. Of the children, about 100 require hospitalization. In a U.S. study it was found that one third of the children admitted to hospital for tap water burns died of their injuries.

Figure 2.1.9 shows the time taken to cause third degree burns in an adult, as a function of water temperature. Children are burned more quickly. It is obvious that water in the temperature range which might frequently be reached in a solar energy system, and which might be no problem from the materials point of view, can be highly hazardous to the user. This is particularly true because of the variation in the temperature of solar heated water. (Most scalds to able-bodied adults occur when they are travelling.)

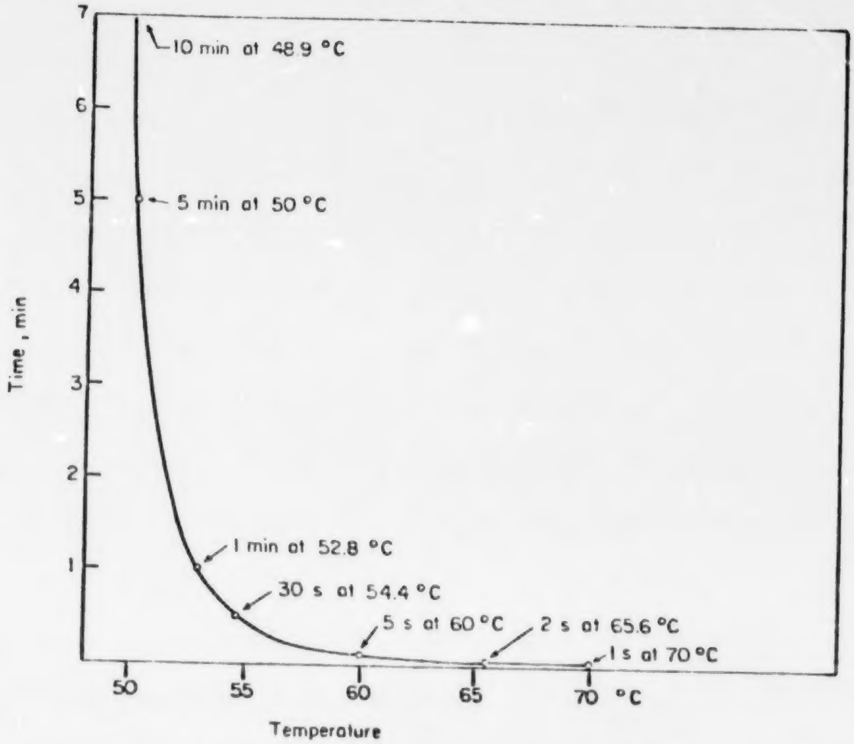


FIGURE 2.1.9

EXPOSURE TIME AND WATER TEMPERATURE
TO CAUSE THIRD DEGREE BURNS

Some plumbing codes require that there must be a storage tank between a tempering valve and the point of use, to eliminate the possibility of short term temperture transients. This requirement can be met and water users can be protected by installing the tempering valve between the tanks of a two tank system. This approach decreases the energy storage capacity of the auxiliary tank, but this is not likely to be a problem when solar energy input is so high that the tempering valve comes into use. In this arrangement the auxiliary heat source may be used to make up auxiliary tank loss even when solar energy is available and not utilized. However, to avoid this it would be necessary to set the tempering valve temperature above the thermostat setting on the auxiliary tank. This would reintroduce the risk of scalding due to unexpected water temperature changes.

The third problem, that of damage to the system by excessively high temperatures, has several possible solutions. One is to design the system to have sufficient heat loss that it never exceeds the safe temperature limit even under no-load conditions. This approach results in excessive losses under other operating conditions.

A second solution is to shut down the circulating pump when the storage tank reaches the temperature limit, and allow the collectors to stagnate. This approach has two disadvantages. One is that glycol solutions degrade faster at high temperatures. The other is that, although collectors should be designed to endure their stagnation temperatures, material degradation will be faster and the lifetime of the unit will be shorter if it stagnates frequently. In drainback and draindown systems, this technique is more feasible, since the glycol degradation problem does not exist.

A liquid-to-air heat exchanger can be used to dump heat from a solar energy system to limit its temperature. It must be installed outdoors since extra heat is definitely not required indoors during the part of the year when the solar energy system will overheat. A good location is on the collector return line, as a parallel branch, isolated by two three-way valves. When the limiting temperature is exceeded, the valves open and the heat exchanger fan is turned on.

The heat exchanger, with a fan, two valves and the associated controls is a very expensive way to dispose of excess heat for a few hours a year. (If the system overheats for more than a few hours then the system is too large relative to the load.) A cheaper approach is to dump water from the storage tank through a thermostatic temperature relief valve. This is a resetting valve, which closes when the temperature of the stored water drops back below the set temperature. An ordinary temperature relief valve, which trips and stays open when the set temperature is exceeded, can not be used for this purpose.

This method of limiting the system temperature wastes water, but the cost will usually be insignificant compared to the capital cost of a cooling coil. The main application of this approach is to provide protection against unexpected overheating due to unexpected loss of load. By definition, this will seldom occur. Expected overheating should be dealt with by decreasing the size of the collector array relative to the load.

The final decision, on whether a cooling coil or the use of mains water is cheapest, must be made by the designer for the specific application. In general the use of mains water will turn out to be the more economical solution to the problem of disposing of surplus heat.

2.2.5 Collector Coupling Devices For Liquid Systems

The pipe connection to the solar collector is a critical area of concern. Fluid leakage at the collector couplings can result due to pipe expansion and contraction, galvanic corrosion and poor workmanship.

When several collectors are employed in a long array the expansion and contraction of the pipe can result in over 3.0 mm of travel for a 3.0 m length of supply and return headers. To accommodate this travel, the piping must be allowed to change in length relative to the panels to which it is connected. In order to minimize the stresses on the collector couplings the connections between the panels and the headers must allow for motion. Three methods are shown in Figure 2.1.10 which allow for external header motion.

Item A shows a steel nipple which is welded to the header with a coupling on a short nipple screwed into the panel. The bend in the nipple assists in absorbing expansion. When up to four collectors are employed the nipple should be approximately 20 cm in length. If larger collector assemblies are used then the nipple length should follow standard piping practice.

Item B shows flare type steel fittings for making the connections to both the collector and the header. An "S" bend, not shown, is desirable

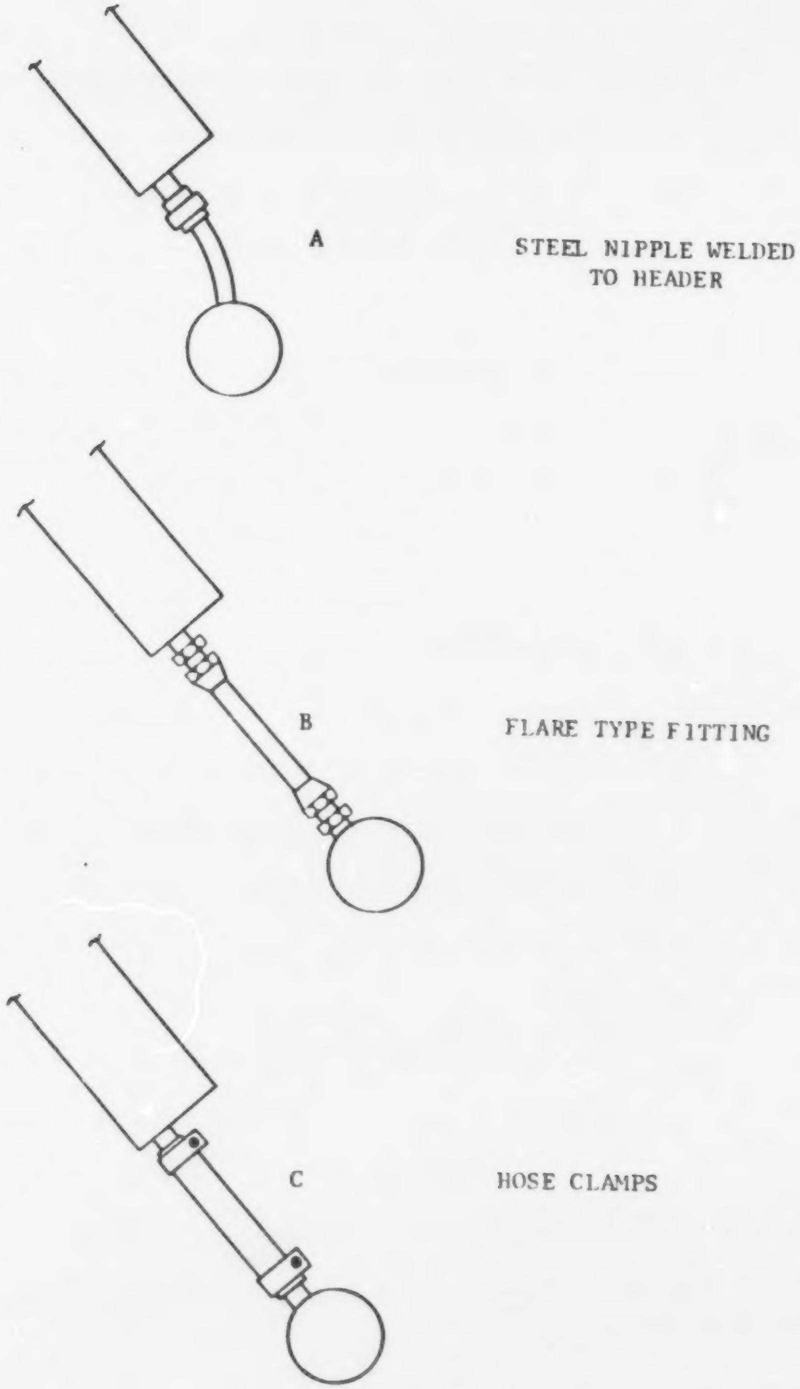

A
STEEL NIPPLE WELDED TO HEADER
B
FLARE TYPE FITTING
C
HOSE CLAMPS

Figure 2.1.10
Flexible Manifold Connections

to allow for expansion and dimensional differences between the header and adjacent collectors.

The third method shown in Item C is the use of an elastomeric hose. Two concerns with respect to hose material are the durability when exposed to sunlight for long period of time and the permanency of clamping. Many hose materials are available but the material selected should be compatible with the fluid used. Silicon based hoses are commonly used as they are compatible with the majority of solar system heat transfer liquids which are combinations of ethylene, glycol and water.

Several types of hose clamps are used with elastomeric hoses. The most common clamp types are the screw, crimp and spring clamp. The screw clamp can slice a hose when overtightened but this can be prevented if a smooth liner is used. The hose may "set" over time and so the clamp will have to be checked to insure clamp tightness. Spring clamps have been successful in avoiding hose damage and in automatically compensating for hose "setting". In general a spring clamp has less clamping force than a screw clamp and there have been cases in which expansion cycling has gradually worked the hose loose. If a shoulder is included on the end of the pipe connection then the spring clamp can prevent this problem from occurring. A crimp clamp is another option, which has not often been used.

The other methods for providing expansion/contraction protection are by using victaulic couplers and expansion bellows. Both of these devices are better suited for use with internal manifolds and for interconnecting panels in vertical arrays and not as direct connections to the collectors. By including these expansion/contraction protective devices on either interior or exterior manifolds the stresses due to expansion/contraction will be reduced and the potential for leakage at the collector couplings is minimized.

A second source of liquid leakage will occur due to corrosion in the pipe system. Galvanic corrosion will occur at couplers if dissimiliar metals are used. As a general rule, the use of dissimiliar metals should be avoided in a hydronic system. If this rule is not followed and appropriate preventive measures not taken, then galvanic corrosion can reduce the lifetime of the solar system.

Several measures can be employed which will prevent galvanic corrosion.

1. Ideally, the same metal should be used throughout the system. Wherever possible, combinations of metals which are close together in the electromotive series should be selected.
2. If dissimiliar metals have to be used then the area of the less noble metal should be relatively large compared to the area of the more noble metal. For example, a small area of steel should be coupled with a large area of aluminum.
3. Galvanic corrosion can be reduced or prevented by slowing the flow of electric current between the two metals by installing electric insulating materials between the two metals.
4. A protective coating such as paint should be applied to the more noble and more corrision-resistent metal.
5. Add chemical inhibitors to the electrolyte. Inhibitors can be classified as either anodic or cathodic depending on what metal is to be protected.

These measures can be taken either separately or in combination.

The last significant cause of leakage at collector couplings is due to poor workmanship. This is a particular concern for soldered connections. To ensure complete penetration of solder into a pipe connection, both the inside of the connector and the outside of the tube should be soldered separately. The connector and tube can then be joined together while the solder is sufficiently soft. By reheating the joint a tight connection will result. Another important consideration when solder is used is that the solder selected be compatible with the pipe material to be soldered. For example, if copper fittings are used then 95-5 tin-antimony solder must be used as ordinary solder will not stand the high temperature that may occur.

If threaded unions are used then additional sealing with Teflon tape should be included. This should be done by tightly drawing the Teflon tape around the threads in such a way that it will not be undone as the fitting is threaded on. When using pipe tape or dope, two or three turns by hand followed by two or three turns with a wrench should secure the joint.

2.3.1 Structural Mounting Details

Each type of collector and each method of installation produces its own characteristics. The collector manufacturer should provide well illustrated mounting procedure guidelines. This section will deal with the various aspects of mounting collectors to sloped and flat roofs.

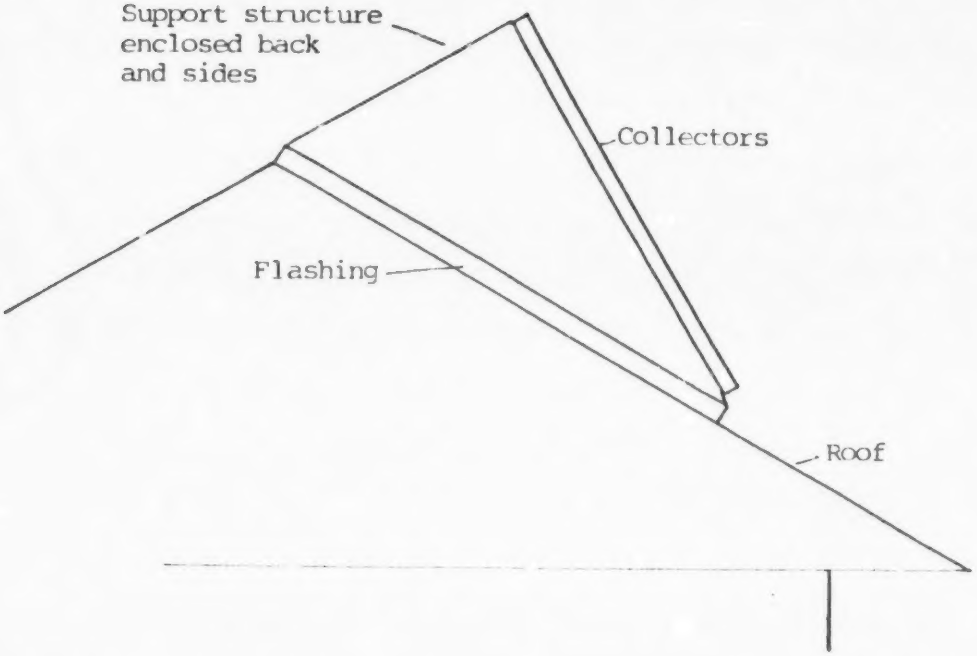
The majority of sloped roof installations will be on small residential houses which have wood as the roofing material used. When a collector is mounted on a sloped roof it can be either mounted directly to the roof or with a stand-off mounting structure if the pitch of the roof is less than the desired pitch of the collectors. The loading requirements are basically limited to the dead load of the collector since the collector installation should not trap wind or snow which would increase the live load. Directly mounted collectors can be attached to the roof in several ways, with the selection dependent on the mounting hardware supplied with the collector. With shingle roofs, the collector support points can be anchored to a plate which spans the roof joists or to wooden sleepers or metal channels which are mounted directly on the roof. When sleepers or metal channels are used, they should not be installed across the slope of the roof or they will create a dam which would retain runoff and so allow water to penetrate the shingles. The sleepers and metal channels should run with the slope of the roof and if roofing cement is applied between each layer of shingles, under the sleeper and between the sleeper and the shingles, leaks should not occur.

When the pitch of the roof does not meet the desired pitch of the collector, the collector should be mounted on a stand-off mounting structure. The design of the stand-off mounting frame will require many of the same considerations that are encountered with flat roofs, these factors will be discussed later. After a stand-off mounting structure has been installed it should be enclosed with weathertight sheathing and have waterproof flashing at the roof surface. This is done to prevent snow from collecting behind the collectors, and because it would be shaded, it would freeze more quickly and thaw more slowly. By providing enclosed back and sides to the stand-off mounting the potential of live loads due to snow accumulation is reduced. A simple illustration of a stand-off collector installation is shown in Figure 2.1.11.

The most common location for the larger solar projects has been on the flat roofs of industrial and municipal buildings. The distance between the collectors vertical supports that are anchored to the roof will be a function of the building's structural capacity to accommodate point loads. The orientation of the collectors to the building structure will also affect the arrangement of the collectors support structure. If the collectors run at right angles to the secondary building structure then the legs of the collector support structure could be located at regular intervals on the secondary building structure. If, however the collectors run parallel to the secondary members then the collector's vertical supports must be located directly over a single secondary member. This results in either the member supporting the total load or in an additional structural frame being required above the roof to carry the collector supports.

Collector support structures can be built from preservative treated wood, or from pipes or angles made of aluminum, iron or steel. The structure should be strong enough to withstand the wind forces, the snow load and the dead weight of the collectors. Structures on flat roofs should be constructed so as to elevate the base of the collector above the average level of snow accumulation.

For collectors mounted on flat roofs a different type of collector support structure and anchoring procedure is required than for sloped roof installations. On smaller buildings with a wooden building structure


Support structure enclosed back and sides
Collectors
Flashing
Roof

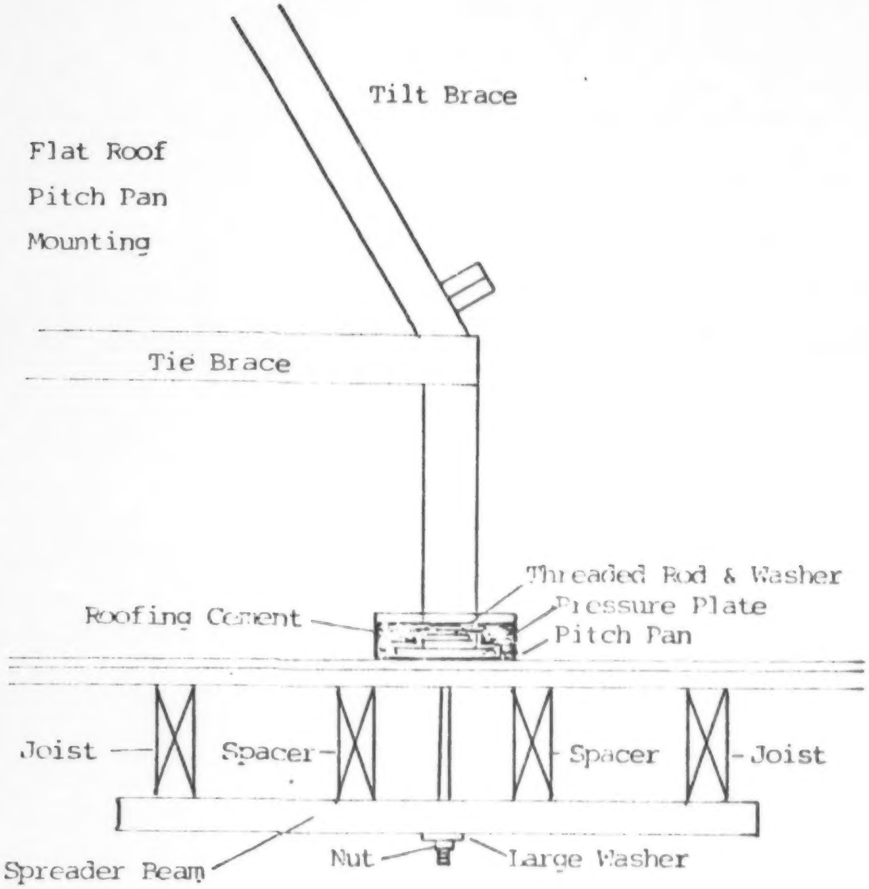
Enclosed stand-off mounting. The enclosed stand-off support structure may be shingled, covered with siding or painted to match the appearance of the building.

FIGURE 2.1.11

a pitch pan mounting can be used, as shown in Figure 2.1.12. Pipe column supports, as illustrated in Figure 2.1.13, should be used on the steel roofs while the curb support shown in Figure 2.1.14 should be used for collectors installed on a concrete roof. When curb supports are used their length should be kept to approximately 1.2 metres so that the curbs do not restrict rainwater movement. The pipe column and curb supports shown in Figure 2.1.13 and 2.1.14 are the recommended support methods given in Public Works Canada publication Structural Design for Solar Collectors, Briefing Document S-2.

A wide variety of collector racks and collector support structures can be used. The simplest collector racks are those employed on small collector installations were only a few collectors are used to form an array. The rack shown in Figure 2.1.15 is typical of the collector racks supplied and recommended by collector manufacturers for a small collector array installation.

When a large collector array is used then typically a long span, triangular shaped structure is required. The most common frame material used is round aluminum tubing, square steel tubing and hot rolled standard steel sections. There is a tendency to select aluminum or steel tubular space frames which results in an aesthetically pleasing design but can be extremely costly and labour intensive. In general, hot rolled steel, wide flanged beams and steel angle structural support systems are less expensive. The dead load of these frames can have a significant impact on the building's structural integrity so that when a large mounting installation is required the appropriate structural experts should be consulted.


Tilt Brace
Flat Roof
Pitch Pan
Mounting
Tie Brace
Threaded Rod & Washer
Roofing Cement
Pressure Plate
Pitch Pan
Joist
Spacer
Spacer
Joist
Spreader Beam
Nut
Large Washer

Typical flat roof pitch pan mounting. The underside of the pitch pan is buttered with roof cement before attachment to roof. After the lag screw or threaded rod is in place the pitch pan is filled with pitch or plastic roof cement. The fastening hardware is then completely covered to prevent water from getting down into the screw threads.

FIGURE 2.1.12

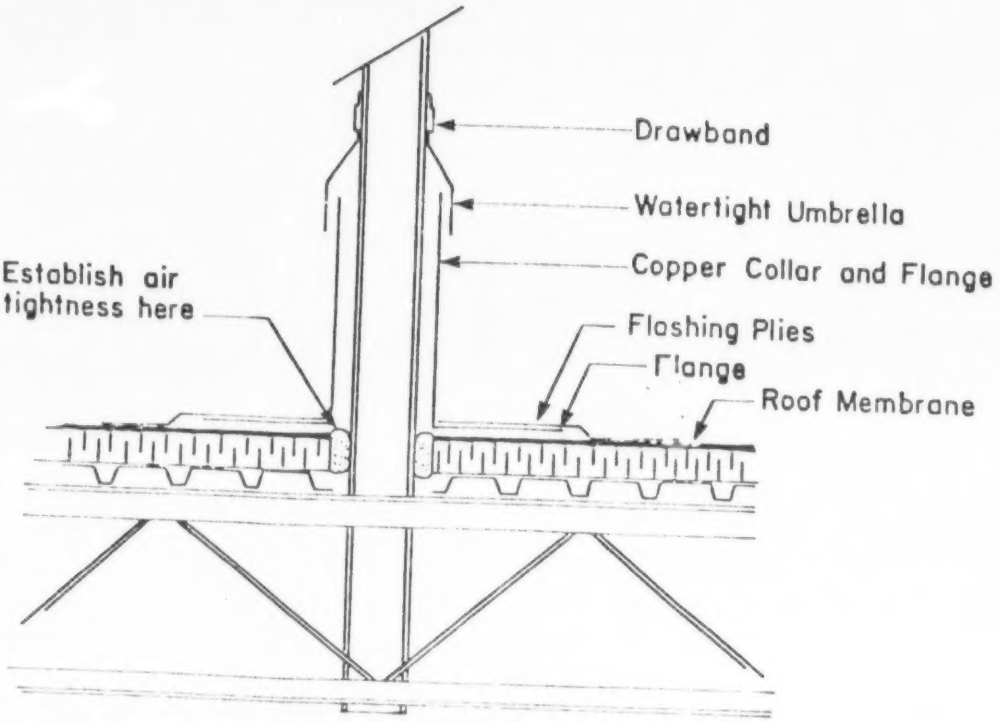

Drawband
Watertight Umbrella
Copper Collar and Flange
Establish air tightness here
Flashing Plies
Flange
Roof Membrane

FIGURE 2.1.13

Pipe column support.

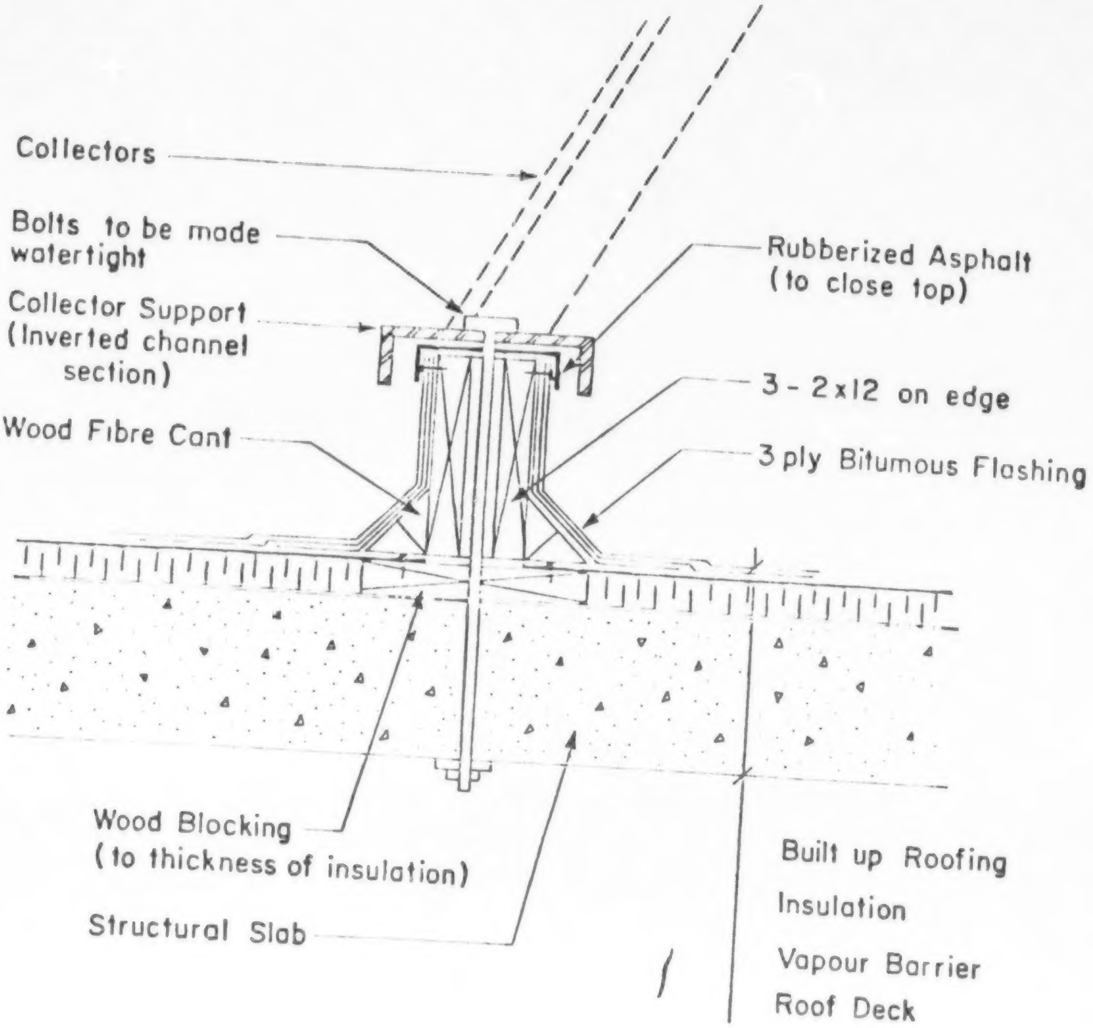

Collectors
Bolts to be made watertight
Collector Support (Inverted channel section)
Wood Fibre Cant
Rubberized Asphalt (to close top)
3 - 2x12 on edge
3 ply Bitumous Flashing
Wood Blocking (to thickness of insulation)
Structural Slab
Built up Roofing
Insulation
Vapour Barrier
Roof Deck

FIGURE 2.1.14

Curb support.

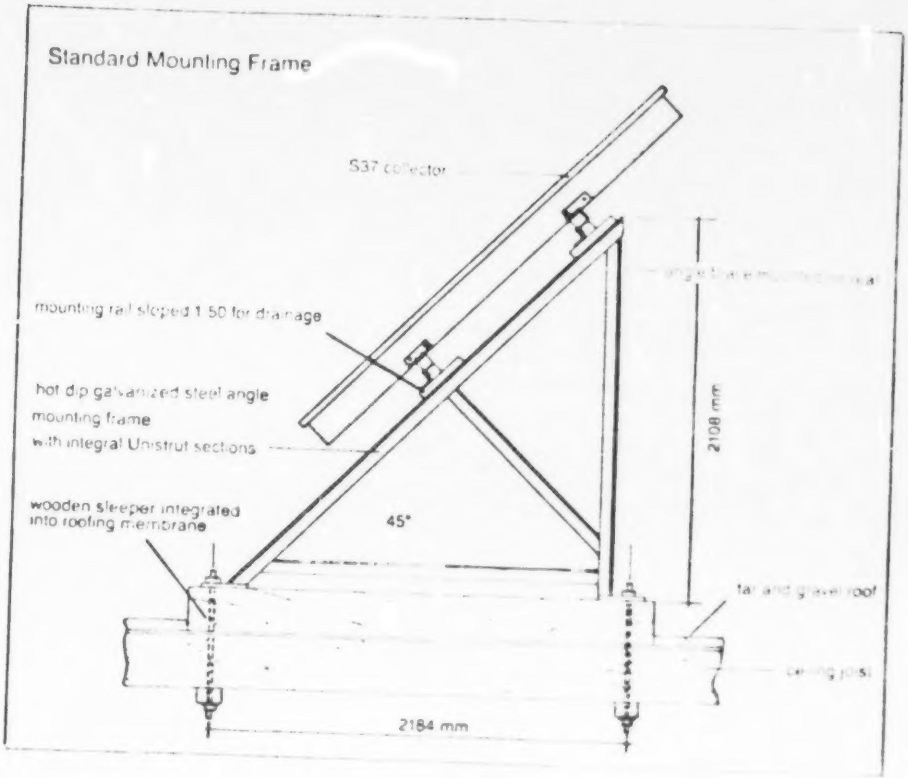

Standard Mounting Frame
S37 collector
angle frame mounted on rear
mounting rail sloped 1:50 for drainage
hot dip galvanized steel angle
mounting frame
with integral Unistrut sections
wooden sleeper integrated
into roofing membrane
45°
2108 mm
tar and gravel roof
ceiling joist
2184 mm

FIGURE 2.1.15

If several rows of collectors are installed one behind the other, then a rear surface sloping towards the next row of collectors should be constructed out of plywood sheathing. If this rear surface is painted white or covered with a reflective film it will reflect radiation onto the next row of collectors. This type of reflector improves performance and also reduces wind lift and wind chill on the row of collectors.

2.3.2 Pipe or Duct Installation and Methods of Protecting It From Weathering

Complete insulation of pipes and valves is essential for successful solar system performance. Unfortunately most of the insulations that designers deal with were not developed for the solar field. Three types of pipe insulation commonly used for solar heating systems are fiberglass, flexible closed cell forms and rigid forms of urethane and isocyanurate foam.

Fibreglass pipe insulation can be bought in both rigid and semi-rigid form. The semi-rigid form allows the fibreglass to be cut to shape for covering valves and fittings. The rigid fibreglass insulation cannot take the flexible shapes that are often required by solar installations. The rigid-fibreglass is good for long runs of piping but it is not suitable for exterior manifolded collector installations. The fact that fibreglass when wet makes a better conductor than insulator necessitates the use of a waterproof covering on exterior applications of fibreglass insulation. Fibreglass pipe insulation can be bought with a watertight and ultraviolet resistant jacket. Additional waterproofing in the form of waterproof weather resistant tape should be used to seal the crack which occurs along the performed fibreglass insulation halves.

Flexible elastomeric (closed wall) insulations are available in very limited thicknesses and so a second layer is required for an improved R value. The flexible insulation is not affected by water as it is waterproof and can be used as its own waterproofing. However flexible elastomeric insulation is affected by the sun's ultraviolet rays. This type of insulation can be painted to protect against ultraviolet deterioration. If it is not painted, serious deterioration occurs within 6 months and failure will occur within a year.

The rigid foam insulation is roughly comparable in cost to rigid fibreglass for the same R value. Rigid foam insulation deteriorates when exposed to the high heat associated with stagnation, to minimize the risk of damage to the insulation a pipe wrap should be employed. Rigid foam is not readily affected by water but a mild acid can be created which will deteriorate the pipe if water is allowed to contact the rigid foam and pipe surface. Some rigid urethane and isocyanurate foams come with a PVC jacket. The PVC jacket offers some resistance to ultraviolet damage but it is advisable to use manufacturer's recommended protective coating for longer service. When sections of rigid insulation with a PVC jacket are glued together they form a waterproof shell.

The majority of pipe insulation suppliers offer various forms of protective jackets. The most common pipe jackets are composed of either fabric, aluminum or vinyl material. The fabric jacket is a glass-fabric and asphalt combination which provides a high melting point but is not waterproof. An aluminum jacket with a underlying moisture barrier provides the best protection against ultraviolet degradation and physical abuse, and is water resistant. The aluminum jacket is recommended for piping systems having a fluid temperature range of between 0 and 400°F. The vinyl jackets are usually a vinyl/aluminum combination and can be used for a wide range of pipe temperatures. Final selection of a protective jacket will be dependent on the expected fluid temperatures in the pipe system and the cost and availability of the insulation jackets.

The level of insulation on pipe, ducts, and tanks in a solar energy system should be optimized in the same way as for any other design job. The only difference is that the designer is usually faced with the problem of trading off capital investment for insulation against annual cost for fuel, while in a solar energy system design, he is trading off alternative capital investments. Thus he does not have to speculate about future interest rates or fuel prices.

For example, on one commercial scale solar service water heating project, the marginal cost of another collector (installed) was \$209.00/m², and the energy gain was 0.67 GJ/a per m², for a cost of \$312.00 per GJ/year. The indoor piping was 40 mm in diameter. The insulation thickness could be increased from 25 mm to 50 mm for \$10/m. This increase in insulation produced a saving of .049 GJ/a per meter, for a cost of \$203.00 per GJ/a saved. Thus the addition of more insulation was a better investment in this case than the addition of another collector. When the same calculation was repeated for an additional 25 mm of insulation the opposite conclusion was reached.

When carrying out such analysis, the fact that some indoor pipe losses reduce heating costs should be taken into account. In the above case the losses were to a mechanical equipment room which required no heating.

In general, it will be found that the optimum insulation thickness on tanks is considerably greater than on indoor pipes, because the cylindrical geometry in the latter case causes a more rapid decrease of energy saving with thickness. Also, the optimum insulation thickness on outdoor pipes will be greater because of the greater temperature difference, although the higher cost of outdoor insulation exerts an influence in the opposite direction.

2.3.3 Ground Reflectance and Its Effect On Thermal Performance

Tables 2.1.3.1 and 2.1.3.2 list the albedo (reflectance in the solar radiation wavelength range) of several kinds of natural ground cover. Table 2.1.3.3 indicates how the albedo changes with incident angel.

Collectors in typical installations may face a wide range of man-made materials. Albedos for some of them are listed in Table 2.1.3.4. Many others can be found in various text and reference books on thermal radiation.

The problem with estimating surface reflectance is that the actual values will vary widely for materials with a single description. For example asphalt pavement is listed as having a reflectance of 0.07, yet gravel is listed at .71, so that real pavements and real roofs can be anywhere between the two.

Luckily, the solar energy system design engineer is usually dealing with a retrofit situation. It is possible to get a good approximation of the reflectance of the materials in the field of view of a set of solar collectors by taking light meter readings at several times during the collector operating day and comparing the brightness with that of a surface of known reflectance on which bright sunshine is falling. The light meter must be appropriately shaded during both sets of readings.

In the field of view of the solar collectors, many may be fully or partially snow covered during the winter. It is then necessary to estimate the fraction of snow cover and the length of the snow season.

Obviously, there are great uncertainities associated with evaluating ground reflectance. However, the overall effect of this number on the performance of solar energy systems is not great. Figure 2.1.20, in the following section shows that the effect of increasing ground reflectance from 0.2 to 0.8 for the entire year is only a 4% increase in the energy gain of a double glazed collector at a 43° tilt, and 6% for a single glazed collector. Therefore the designer who makes the best possible estimate of ground reflectance on a month by month basis is not likely to introduce a major error into his FCHART prediction if his reflectance estimate is slightly in error.

TABLE 2.1.3.1

TYPICAL VALUES OF ALBEDO

Surface	Albedo, %
Snow, fresh	70-95
Snow, several days old	70
Dry dune sand	37
Moist dune sand	24
Wet grass in sun	33-37
West grass, no sun	14-26
Dry grass	15-25
Forest: pine, fir and oak	10-18
Ocean	2-7

TABLE 2.1.3.2

LATITUDINAL AND CONTINENTAL MEANS OF SURFACE ALBEDO OVER NORTH AMERICA

Latitude Zone (°N)	Prevailing Surface Type	Continential Surface Albedo (percent)				
		January 31			March 31	Summer
		Average Snow Cover	Heavy Snow Cover	Light Snow Cover	Average Snow Cover	
65-70	Tundra	82.8	82.8	82.7	82.8	16.1
60-65)	Tundra and Conifer forest	(67.3	67.3	58.2	67.3	15.6
55-60)		(59.1	59.1	54.8	57.7	16.5
50-55)	Forest and grassland	(50.3	50.3	45.8	48.0	14.6
45-50)		(46.4	48.9	28.4	37.6	14.8
40-45)	Shrubland, cropland and woodland	(37.9	50.4	19.0	30.5	15.8
35-40)		(28.5	40.8	16.0	21.1	16.5
	Continental mean	43.0	47.4	34.7	39.4	16.0

TABLE 2.1.3.3

ALBEDO OF A GRASS SURFACE (IN PERCENT) AT TORONTO

Hour (true solar time)	07	08	09	10	11	12	13	14	15	16	17
Sept. 9, 1961 (clear)	25.6	22.3	22.5	21.3	20.8	20.6	20.8	21.4	22.8	24.2	25.3
Sept. 26, 1961 (cloudy)	19.7	19.1	20.2	17.5	19.4	17.6	18.1	19.3	20.0	19.7	28.6

TABLE 2.1.3.4

ALBEDOS OF SEVERAL MATERIALS

Tarnished copper	0.36
Asphalt pavement	.07
Red brick	.25
Clay	.61
Concrete roofing tile:	
uncoloured	.27
brown	.09
black	.09
Bare earth, plowed	.62
Gravel	.71
White marble	.54

2.3.4 The Incident Angle Modifier

The incident angle modifier is a correction factor introduced to account for the variation of $(\alpha\tau)$ with the angle of incidence of the radiation falling on the collector. It is defined as

$$K_{\alpha\tau}(\theta) = (\alpha\tau)_\theta \,/\, (\alpha\tau)_n$$

where $(\alpha\tau)_\theta$ is the value of $(\alpha\tau)$ measured at the angle of incidence θ and $(\alpha\tau)_n$ is the value measured at normal incidence.

ASHRAE Standard 93-77 specifies the method of measurement of the incident angle modifier. It can be determined in an indoor or an outdoor test, and is found by making efficiency measurements at four different incident angles with the collector coolant inlet temperature equal to the ambient temperature.

Figure 2.1.16 shows typical curves of incident angle modifier as a function of incident angle for three flat plate collectors.

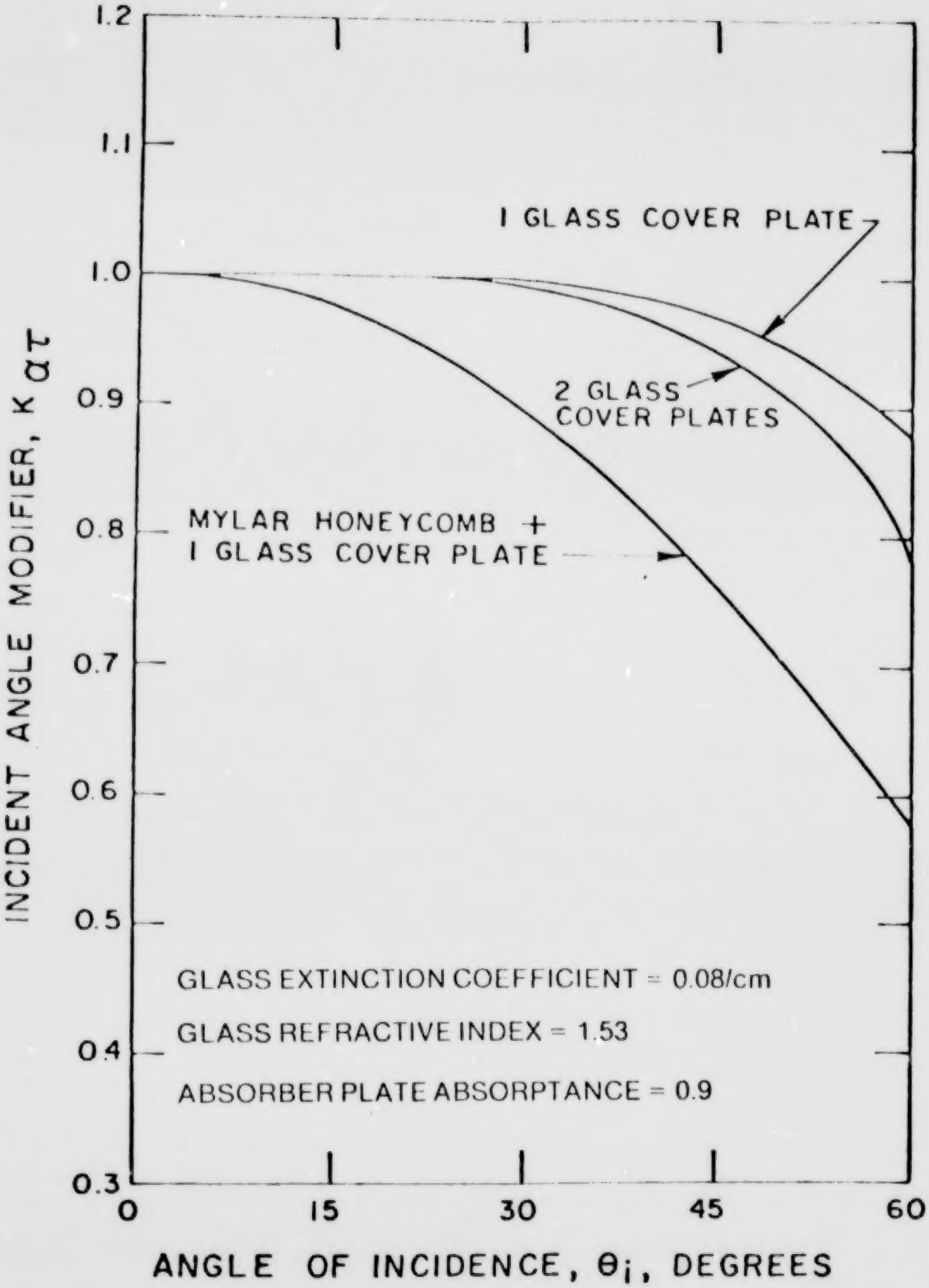
It has been found that the variation of the incident angle modifier can be made linear if it is plotted against $\frac{1}{\cos\theta} - 1$ instead of against θ. Figure 2.1.17 illustrates this method of presentation.

This formulation is convenient for computational purposes because it allows the incident angle modifier to be expressed as

$$K_{\alpha\tau} = 1 + b_0\left(\frac{1}{\cos\theta} - 1\right)$$

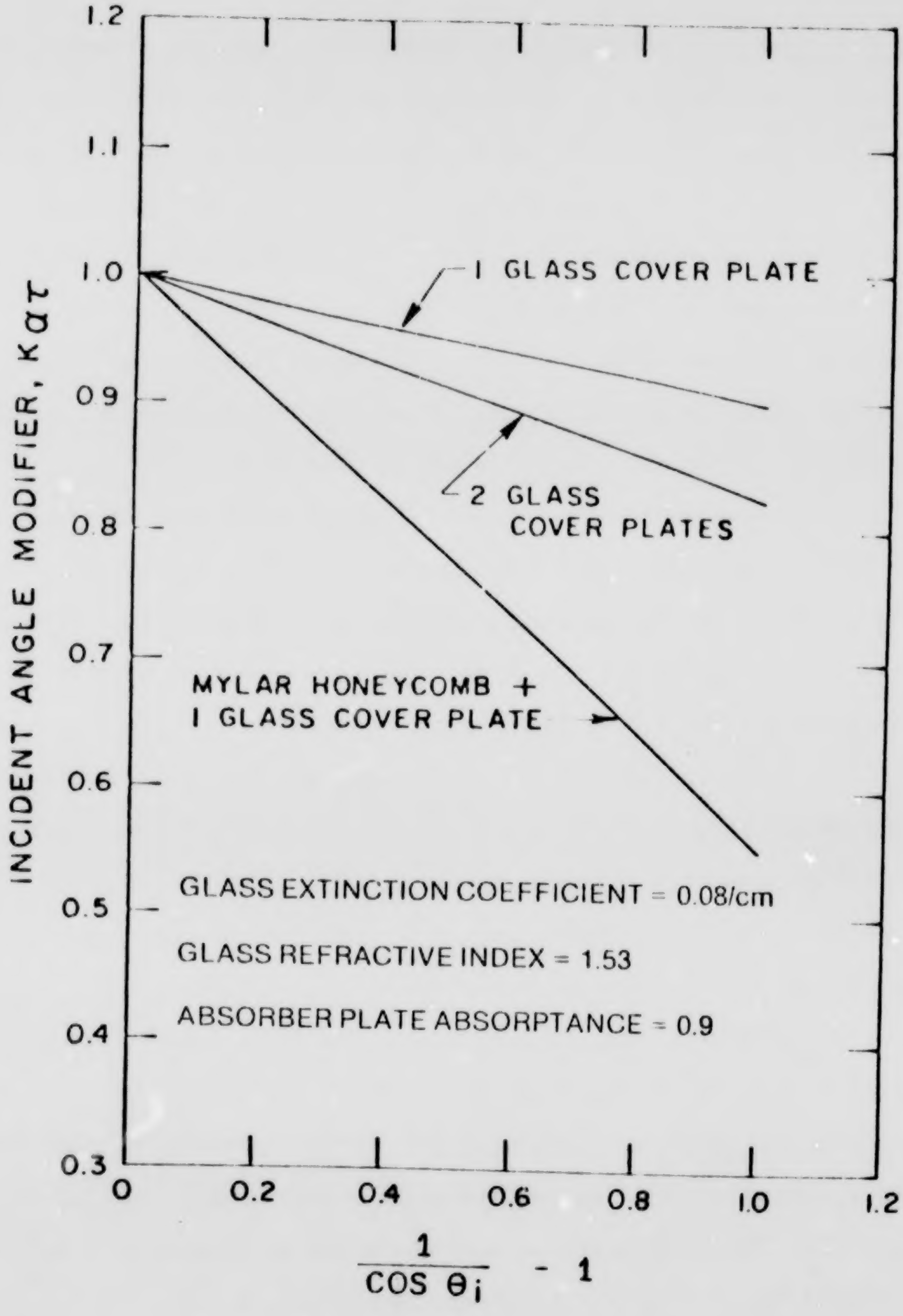
and a single number, the incidence angle modifier coefficient, b_0, is sufficient to define the value of $K_{\alpha\tau}$ for all incident angles.

At first glance, the reduction in collector efficiency caused by non-normal incidence does not appear serious. At 45°, a double glazed



Incident Angle Modifier for Three Flat-Plate Solar Collectors with Non-Selective Coatings on the Absorber

FIGURE 2.1.16



Incident Angle Modifier for Three Flat-Plate Solar Collectors with Non-Selective Coatings on the Absorber

FIGURE 2.1.17

collector loses only about 6%, and at 60°, only about 22%. However, it should be remembered that these corrections are made by subtracting from the efficiency at normal incidence, not by multiplying. Thus, if the efficiency of a certain double glazed collector at normal incidence and with the inlet temperature equal to the ambient temperature is 72%, the efficiency at an incident angle of 60% is 72% x 78% = 56%, and the efficiency at all other values of $(T_{f,i} - T_a) / GT$ is also decreased by an absolute amount of 14%, not by 14% of what it would have been at normal incidence. This point is illustrated in Figure 2.1.18.

Two other factors make the off-normal performance of a solar collector significant. One is the sun's declination. For a collector tilted at 45°, the incident angle will be greater than 20° even at noon, both in mid-summer and in mid-winter. This means that very little collector operating time will be spent in the range of incident angles for which $K_{\alpha\tau} = 1$. The second factor is that diffuse radiation is received over a wide range of angles of incidence. Figure 2.1.19 shows the effective incidence angles for isotropic diffuse radiation from sky and ground as functions of collector slope. The bands shown cover a wide range of single and double glazed flat plate collectors.

An examination of Figure 2.1.19 shows that an effective incidence angle of about 58° is a good approximation for sky diffuse radiation over a wide range of collector tilt angle. For diffuse radiation from the ground, the variation of effective incident angle is much greater than for sky radiation. To obtain a precise estimate of the performance of a collector tilted at 30° or more it will be necessary to use the data from this figure

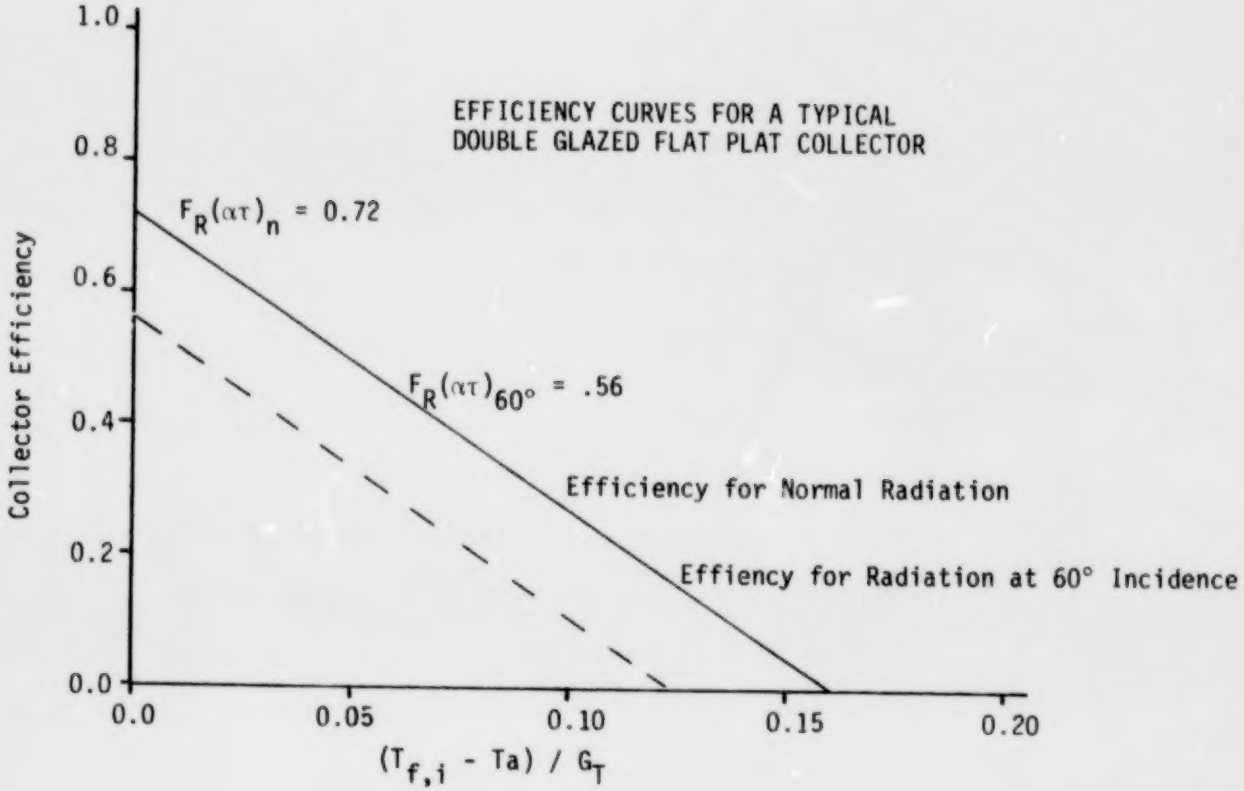


FIGURE 2.1.18

EFFECT OF INCIDENT ANGLE ON EFFICIENCY

along with the incident angle modifier. This is particularly true during the winter, when the ground is brighter than the sky and the accuracy of ground reflected radiation calculations is more important.

When FCHART is used by the design engineer to analyse the performance of a solar energy system using flat plate collectors, he can either supply the computer with the measure value of the incident angle modifier coefficient, b_o, or he can allow the value to be estimated from the number of covers, the index of refraction, the extinction coefficient and the thickness of the covers. In the latter case, the value of $K_{\alpha\tau}$ for each incident angle, θ, is determined from

$$K_{\alpha\tau} = \frac{\alpha \cdot \tau}{[1-(1-\alpha)]\rho d} \cdot \frac{1}{(\tau\alpha)_n}$$

with the reflectance and absorbtance of the covers calculated from the Fresnal equation and Bouger's laws for the value of θ being considered. The variation of the absorbtance of the absorber plate with θ is estimated from an empirical fit to data measured for blackened surfaces.

To calculate $K_{\alpha\tau}$ for diffuse radiation from the sky, the same method as for beam radiation is used, with an assumed incident angle of 60°. For ground reflected radiation the equation

$$\bar{\theta}_r = 89.8° - .57883\beta + .002693\beta^2$$

is used to determine the effective incident angle. This equation fits the curve of Figure 2.1.19.

Evacuated tube collectors differ markedly from conventional flat plate collectors in their response to non-normal radiation. They are not isotropic, and they do not conform to the usual incident angle modifier

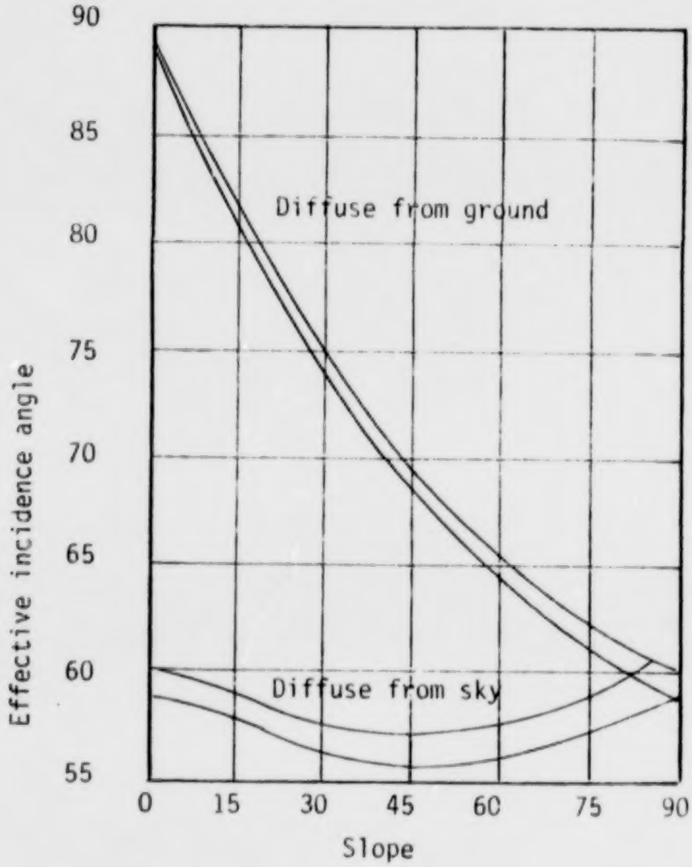


FIGURE 2.1.19
EFFECTIVE INCIDENCE ANGLE OF ISOTROPIC DIFFUSE SOLAR RADIATION

equation. In fact, $K_{\alpha\tau}$ may rise above 1.0 for non-normal incidence. Neither the ASHRAE 93-77 test procedure nor the FCHART computer program deal adequately with this type of collector. The manufacturers have made protests to the ASHRAE 93-77 Revision Committee, and it is possible that a new method of dealing with this problem will be included in the revised version of the standard.

The collector incident angle modifier equation is also inadequate to deal with concentrating collectors of either the imaging or compound parabolic type.

Figure 2.1.20 indicates the impact of the incident angle modifier constant, b_0, on the solar fractions predicted by FCHART. The points marked on each curve are typical values of b_0 for single and double glazed collectors respectively. The range of values of b_0 shown is far greater than could actually occur. As can be seen from the graphs, variations of b_0 of ±50% cause about ±2% changes in the amount of energy collected by a single glazed collector and ±4% changes for a double glazed collector. As would be expected, a very high value of ground reflectance makes a collector more sensitive to variations in b_0.

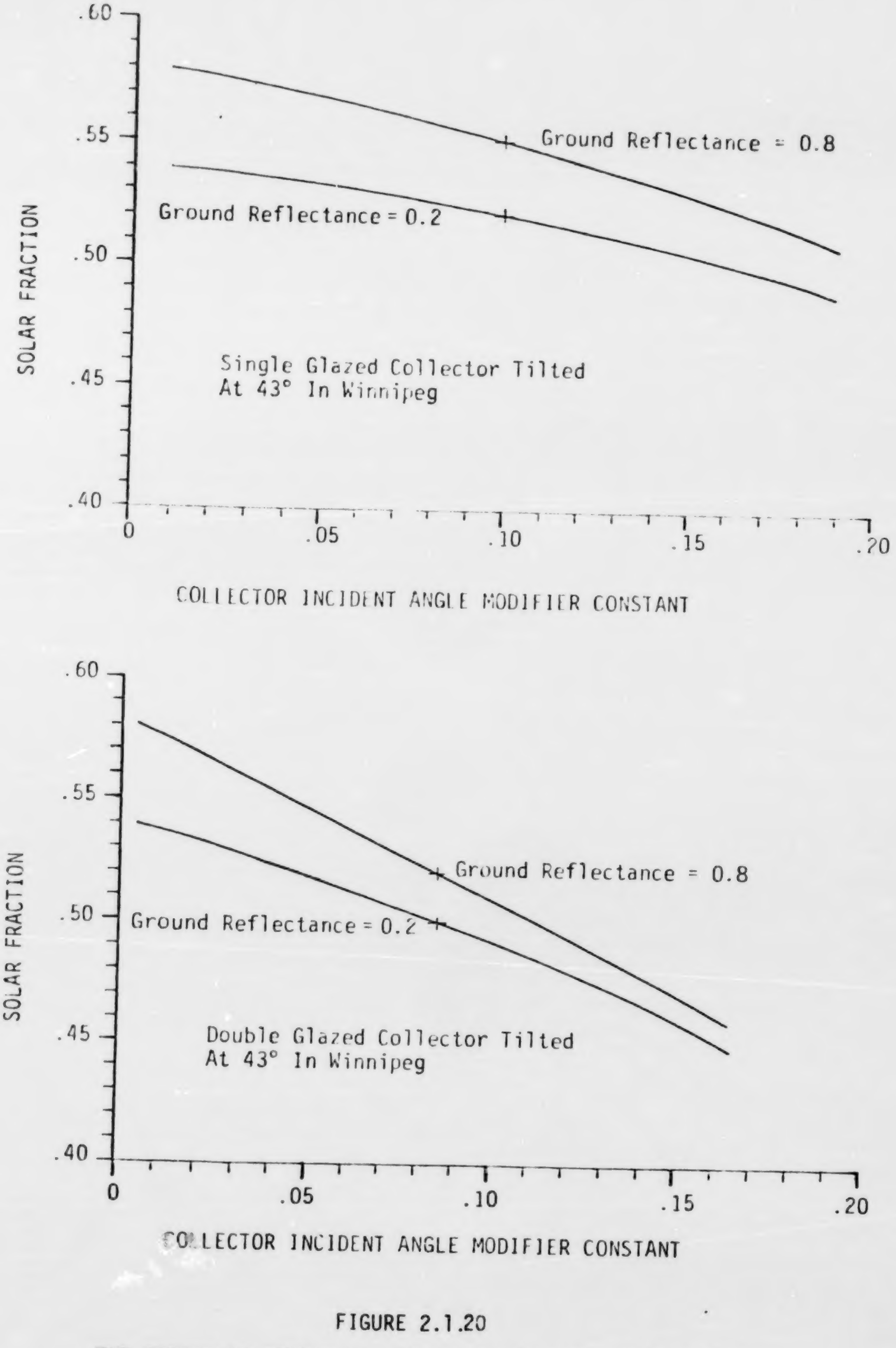


FIGURE 2.1.20

THE EFFECT OF SINGLE VS. DOUBLE GLAZING ON SOLAR FRACTION

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CHAPTER 3

DESIGN APPROACH

3.1 INTRODUCTION

The topics covered in this chapter are:

1. Design Sequence
2. Design Objectives
3. Thermodynamic factors
4. Economic factors
5. Synthesizing these factors into a system design.

In this chapter some of the fundamental concepts that help the designer to develop a successful solar energy system design will be identified. No attempt will be made to deal with all the detailed problems of component design faced by the engineer. Rather, an overall perspective will be taken, and the general approaches that enable the system designer to produce a successful product will be identified.

3.2 DESIGN SEQUENCE

The design of a solar energy system depends on the use to which the energy collected is to be put, and on the physical environment in which the system must operate.

Criteria used in reviewing the potential applications of solar energy in a facility are:

1. the grade of energy necessary to meet the load,
2. the size of the load,
3. annual load profiles,
4. daily load profiles,
5. the convenience of the load,

The first and most important of these factors is the grade of energy (i.e. the load temperature) required for the application. High temperature applications will have high costs per unit of collector area installed and low operating efficiencies. Conversely, low temperature applications can utilize low cost components and achieve high efficiency. For example, a ventilation air preheat system may have unit energy costs of an order of magnitude lower than a space heating system delivering an equal amount of energy.

The size of the load is also very significant. There are many fixed or semi-fixed costs associated with a solar energy system regardless of its size. A large load justifies a large system. A large system provides a larger base to support these fixed and semi-fixed costs. A solar energy system meeting a moderate fraction of a load will operate at relatively high efficiencies when compared to one meeting a large fraction of the same load. A large load presents the opportunity of installing a solar system large enough to enjoy economies of scale, while small enough, relative to the load, to operate efficiently.

Daily and annual energy application load profiles should be compared to solar availability. High summer loads and low winter loads occurring during daylight hours are the ideal. Residential space heating, with large night winter loads and no summer loads is on the negative side of the spectrum while beach-shower facilities are on the positive. The daily load profile is an indication of what the thermal storage requirements will be. Thermal storage is often a major component in system cost, so systems that have no storage may show cost advantages in certain applications over those with storage.

The convenience of an application refers to the ease with which a solar energy system can be integrated into a thermal system. During the design stage, it is relatively inexpensive to modify a structure to support a solar energy system for a particular load application. In an existing structure, the most logical solar application may be too inconvenient and thus too costly to consider.

Once a building has been analysed and the "best" solar application selected, the designer should begin to conceptualize the layout of the system components. Rules of thumb tempered by physical constraints (such as the mounting area available for collectors) are used to select a collector area. Building-operator imposed constraints may further restrict the latitude of designer decisions.

In new construction, a building design can be modified to conveniently incorporate a solar energy system. Retrofit systems involve compromise and trade-offs. However, this does not necessarily mean the system will be much less efficient than an optimally located system. Variations of ±15° from ideal slopes and up to 45° from ideal collector azimuth angles frequently have very little effect on system performance, while allowing the designer to utilize existing building features to reduce system cost. The designer should carefully assess the building's features when laying out system components, and attempt to simplify installation and operation of the system. Collector support structures, ducting and piping and thermal storage can account for a major feature of first costs. Working with, rather than around, building features can result in significant cost savings. As capital costs are the most significant part of the cost of solar energy, they should be controlled as much as possible.

Although capital costs should be kept low, this should not be done at the expense of unit energy costs and reliability. Some low cost systems experience a lot of down time. The cost of repairs and the smaller solar fraction met by the system (due to the unavailability of the system) may result in higher energy costs over the life of the system. Capital and operating costs for the various loads and system configurations should be compared to system performance. The most important parameter in selecting a solar energy system size and configuration is the $/GJ cost of useful energy collected. The designer should select a system that meets all the necessary design criteria while minimizing unit energy costs. This will not usually be the most efficient system possible for the application, but the system should be designed to operate at a high relative efficiency.

There are two areas of solar energy system design that may require analysis by a structural engineer. Roof mounted collectors may not present significant loads on the structure, by themselves. However, wind loads on the array or snow loads that accumulate on a roof because of the "snow fence effect" of the collector arrays may exceed structural capabilities of the building. Public Works Canada, Design/Construction Branch has issued

"Briefing Document S-2, Structural Design Solar Collectors" on the topic of roof mounted solar arrays. The high mass of thermal storage required for some solar energy systems may present structural problems. Generally, thermal storage must be located on structural floors or on slabs of grade.

Every solar energy project is unique and should be approached with an open mind. The solution to each job requires its own mix of economic, technical, space, reliability and aesthetic considerations. The designer should look for opportunities in the structure and the building energy requirements that will yield the lowest solar energy costs possible for the application.

3.3 DESIGN OBJECTIVES

The important objectives of a solar energy system designer are:

1. safety
2. reliability
3. durability
4. maintainability
5. performance/cost ratio

Safety is not an insignificant issue in solar energy systems. Low pressure hot water systems have killed many people and blown buildings apart when controls and relief valves failed. Professional engineers take on serious responsibilities when they design systems, and no engineer who is not qualified and experienced in the design of solar energy systems or other systems of a similar nature should undertake such a design. An engineer should carefully review the design, imagining each control element has failed and each valve incorrectly adjusted, to check for possible unrelieved overpressure or overheating situations. As discussed earlier, contamination of potable water is another potential hazard which must be guarded against.

When in doubt on a safety issue, a design engineer should imagine testifying at an inquest to justify or explain a decision.

Reliability, durability and maintainability are really part of system economics, but are discussed separately because it is so difficult to quantify their cost effectiveness. Experienced design engineers will have developed their own standards on these issues and will continue to get feedback from clients when they design systems that fail in these respects. On the other hand, few clients notice when a system is too expensive because the engineer put in a couple of unnecessary isolating valves, or wrote an unrealistic specification for a control component. A good engineer must monitor the design to ensure that systems are not overdesigned at the client's expense.

Codes and standards are one of the bases of design. Some important references which may be useful in designing solar energy systems are:

1. Government of Canada Master Construction Specification -

   Section 15643 - Flat Plate Solar Collector Panels - 1979-12-31.

2. ASHRAE Standards
   93-77 Flat Plate Collectors (presently under revision)

   95-81 Packaged Domestic Water Heaters

   96-80 Unglazed Collectors

(These ASHRAE standards are all standard methods of testing thermal performance)

3. CSA Standards

F378-1982 - Solar Collectors

F379-M1982 - Solar Domestic Hot Water Systems (Preliminary)

(These CSA standards include thermal performance tests and also requirements for safety and durability)

The solar domestic hot water system standards provide a good starting point for designers of larger solar service water heating systems.

In addition to the design objectives listed above, particular projects may have additional objectives such as system development, training and experience, or performance monitoring. The PUSH program has as its overriding objective the establishment of a Canadian solar equipment industry. Designers of PUSH funded projects must keep this in mind when making cost-effectiveness trade-offs between investment in solar collectors and investment in conventional hardware which makes up the "balance of plant".

3.4 SYSTEM OPERATING RANGES AND COLLECTOR THERMAL PERFORMANCE

The thermal efficiency of a solar collector is a function of the ratio

$$(T_{f,i} - T_a)/G_T$$

where $T_{f,i}$ = entering fluid temperature

T_a = ambient temperature

G_T = insolation on the collector

This number will vary during a typical year from infinity almost to zero, and in some applications to well below zero. To select the right collector the designer must match the collector performance curve to the curve of the frequency distribution of $(T_{f,i} - T_a)/G_T$.

Figure 3.3.1 shows two distributions of this ratio, for inlet temperatures of 5°C and 35°C using Winnipeg weather data and a south facing collector tilted at 45°. (In the real system, the inlet temperature normally varies too, but constant values have been used here to simplify the discussion.) Figure 3.3.2 shows two typical collector efficiency curves, for an unglazed plastic collector and for a glazed collector with a metal absorber.

By comparing the frequency distribution of $(T_{f,i} - T_a)/G_T$ with the collector efficiency curve, a distribution curve for the variation of collector efficiency throughout the year can be produced. Such a curve is shown in Figure 3.3.3, for the glazed collector with an inlet temperature of 35°C.

The curve in Figure 3.3.3 was produced from the curves of Figures 3.3.1 and 3.3.2 by the following means. For each valve of $(T_{f,i} - T_a) / G_T$ on Figure 3.3.1 the correponding frequency on the curve was 35°C inlet temperature was determined. For the same value of $(T_{f,i} - T_a) / G_T$ in Figure 3.3.2, the efficiency of the glazed collector was determined. Then each collector efficiency was plotted against the corresponding frequency in Figure 3.3.3. (For example, the dotted lines on Figures 3.3.1 and 3.3.2 show that, at $(T_{p,i} - T_a) / G_T$, the frequency of occurrence is 310/year, and the efficiency is 27%. This point is illustrated on Figure 3.3.3).

A curve of the frequency distribution of efficiency is of little use in the detailed design of a system, but it does have value in the concep-

FIGURE 3.3.1

Frequency Distribution of $(T_{f,i} - T_a)/G_T$

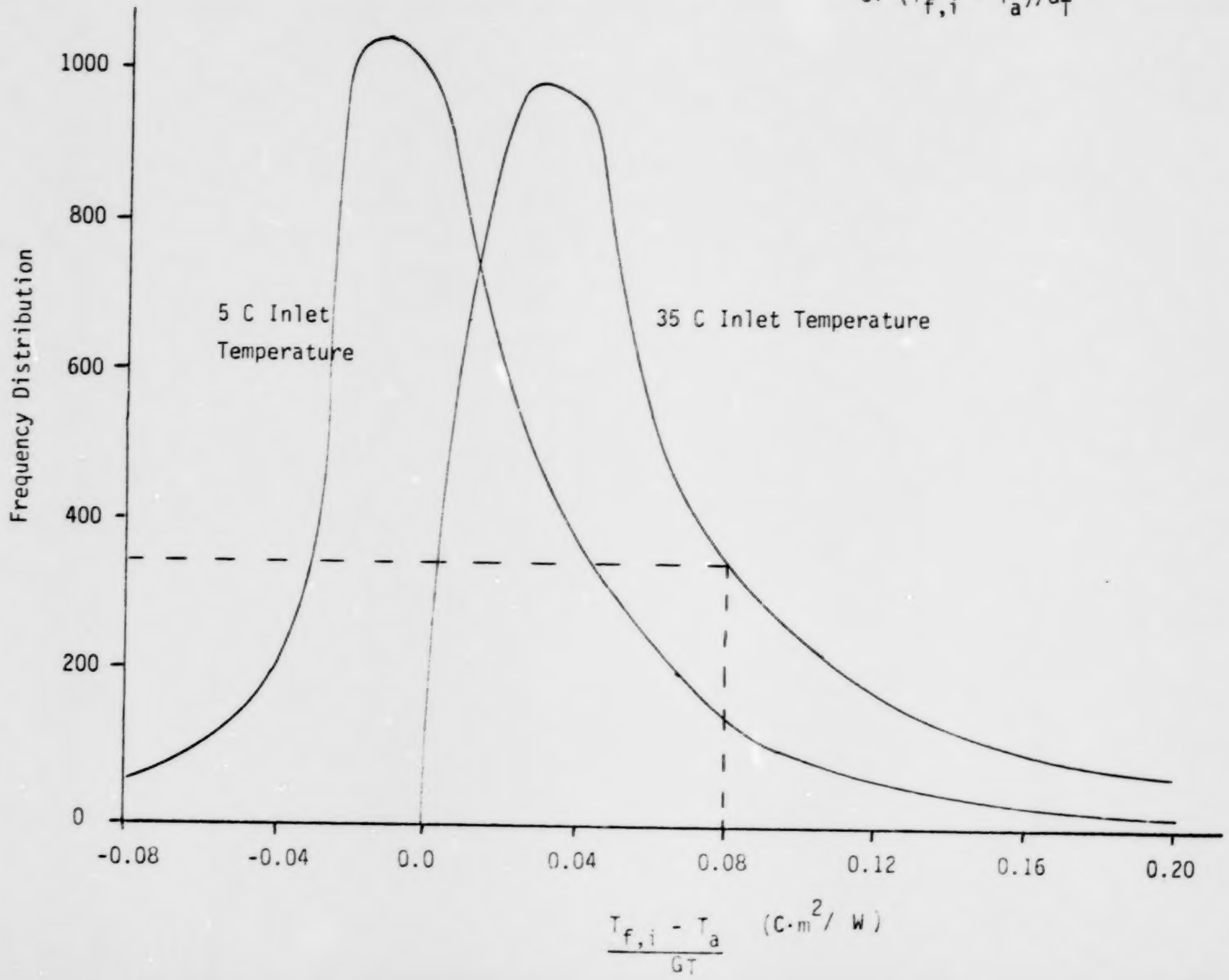


FIGURE 3.3.2

Efficiencies of Two Typical Collectors

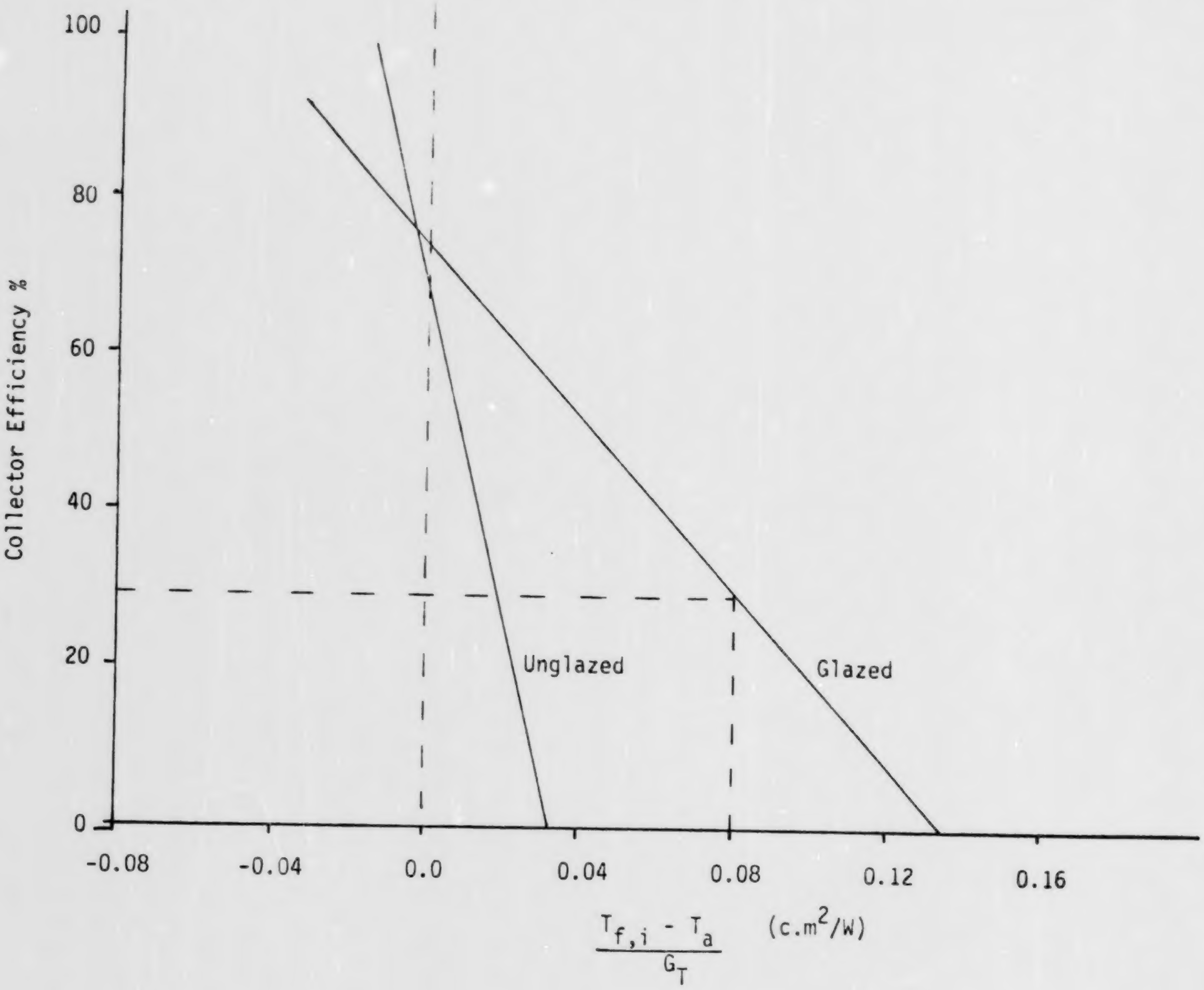


FIGURE 3.3.3

Frequency Distribution of Efficiency

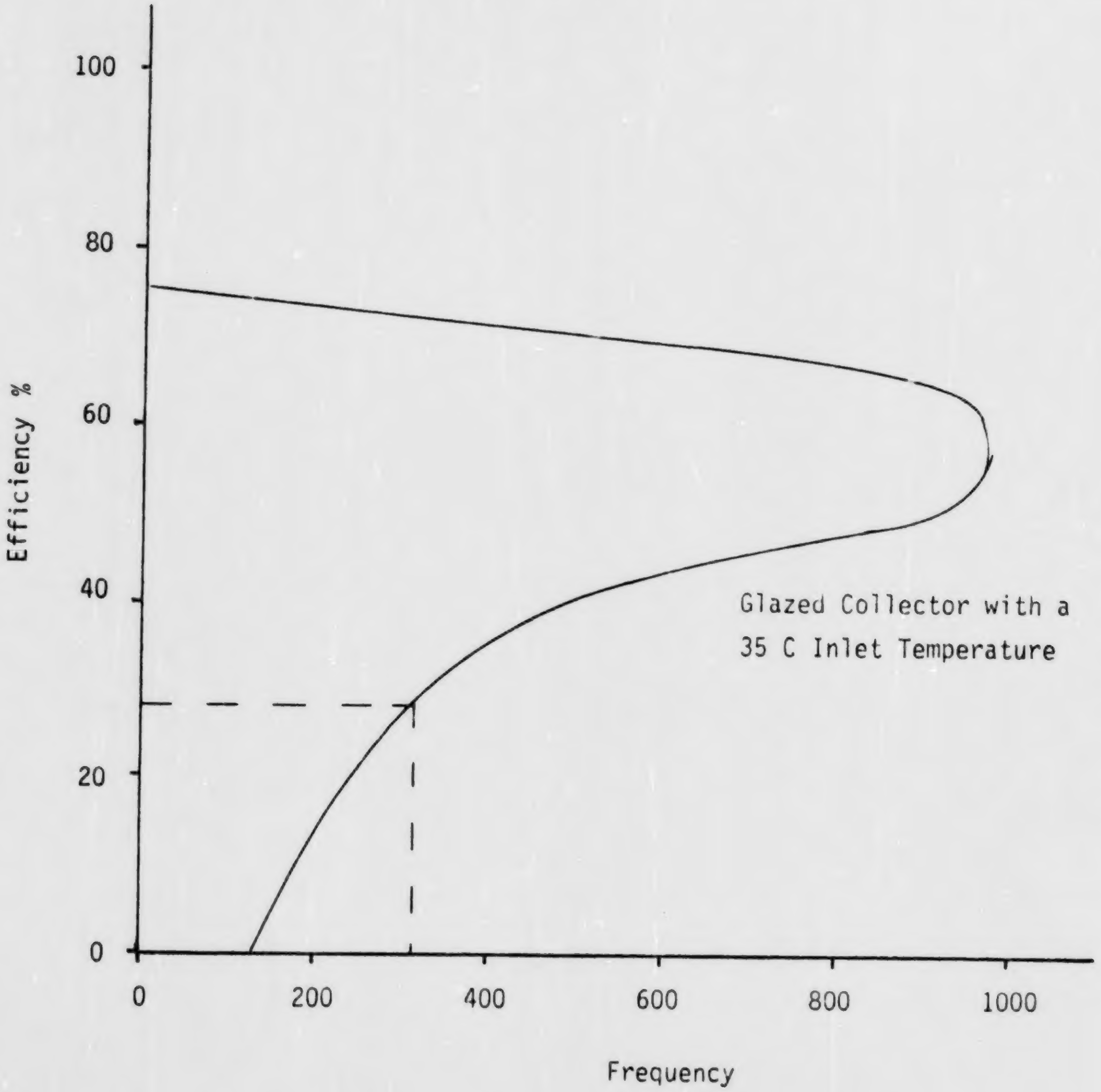


FIGURE 3.3.4

Frequency Distribution of Efficiency
for a 5 C Inlet Temperature

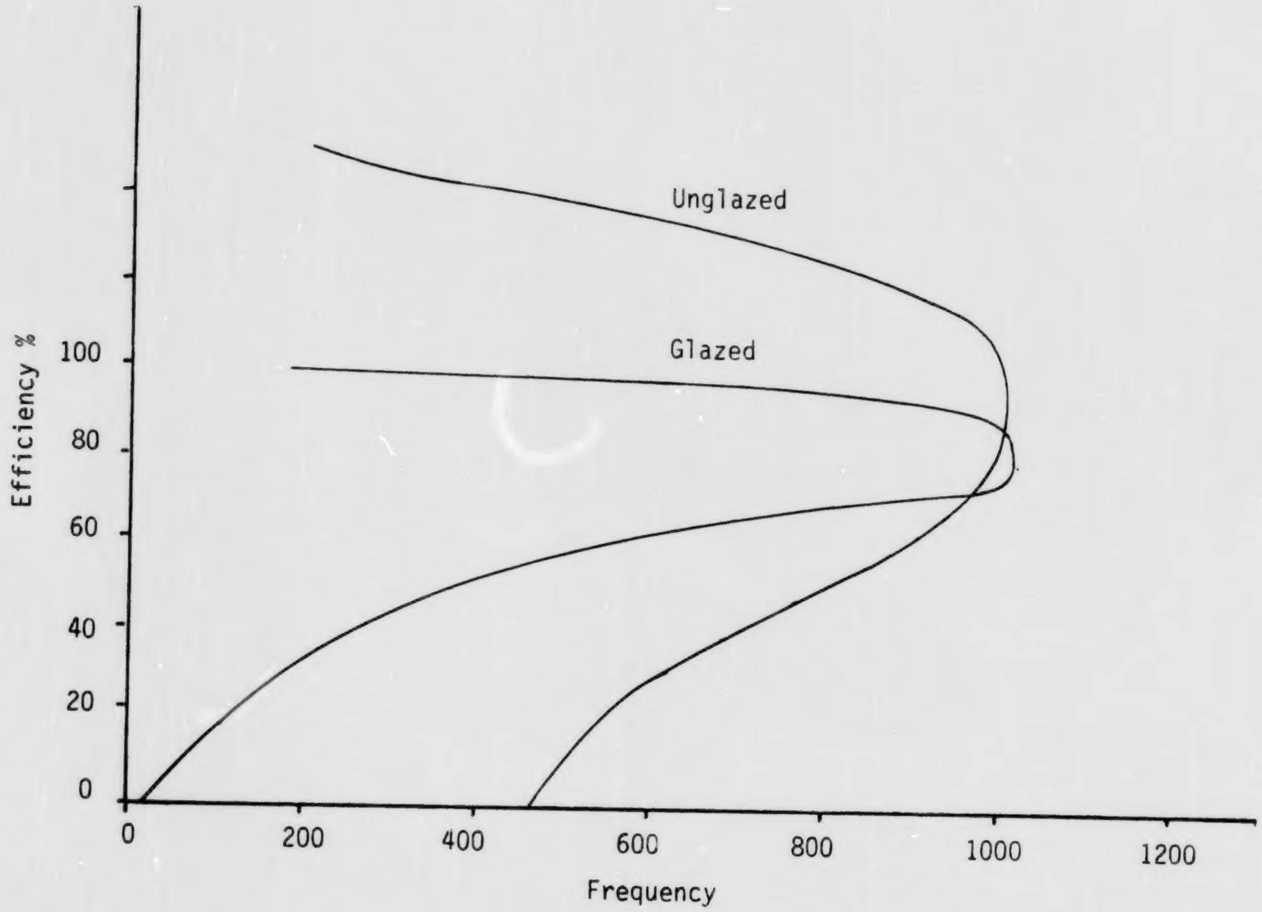
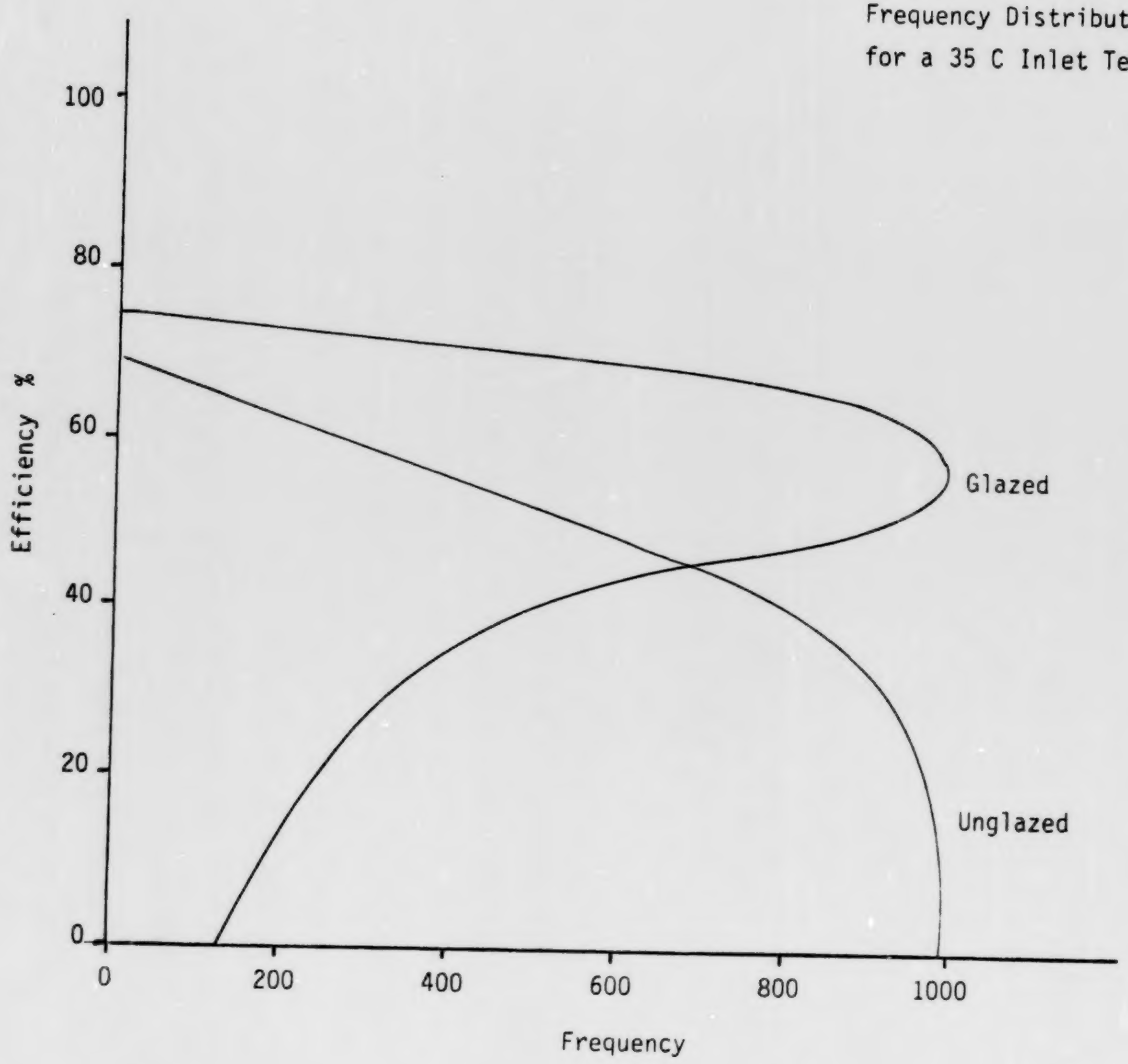


FIGURE 3.3.5

Frequency Distribution of Efficiency
for a 35 C Inlet Temperature



tual design stage, and in arriving at an appreciation of why a particular collector may be best in a given situation. For example, to find the collector with the best range of thermal efficiencies for a given job, a comparison can be made between the efficiency distributions of those collectors being considered. In Figure 3.3.4 and Figure 3.3.5, the two collectors are compared for each of the two loads. As would be expected, the unglazed collector has the best range of efficiencies when the inlet temperature is low and the glazed collector is best when the inlet temperature is higher.

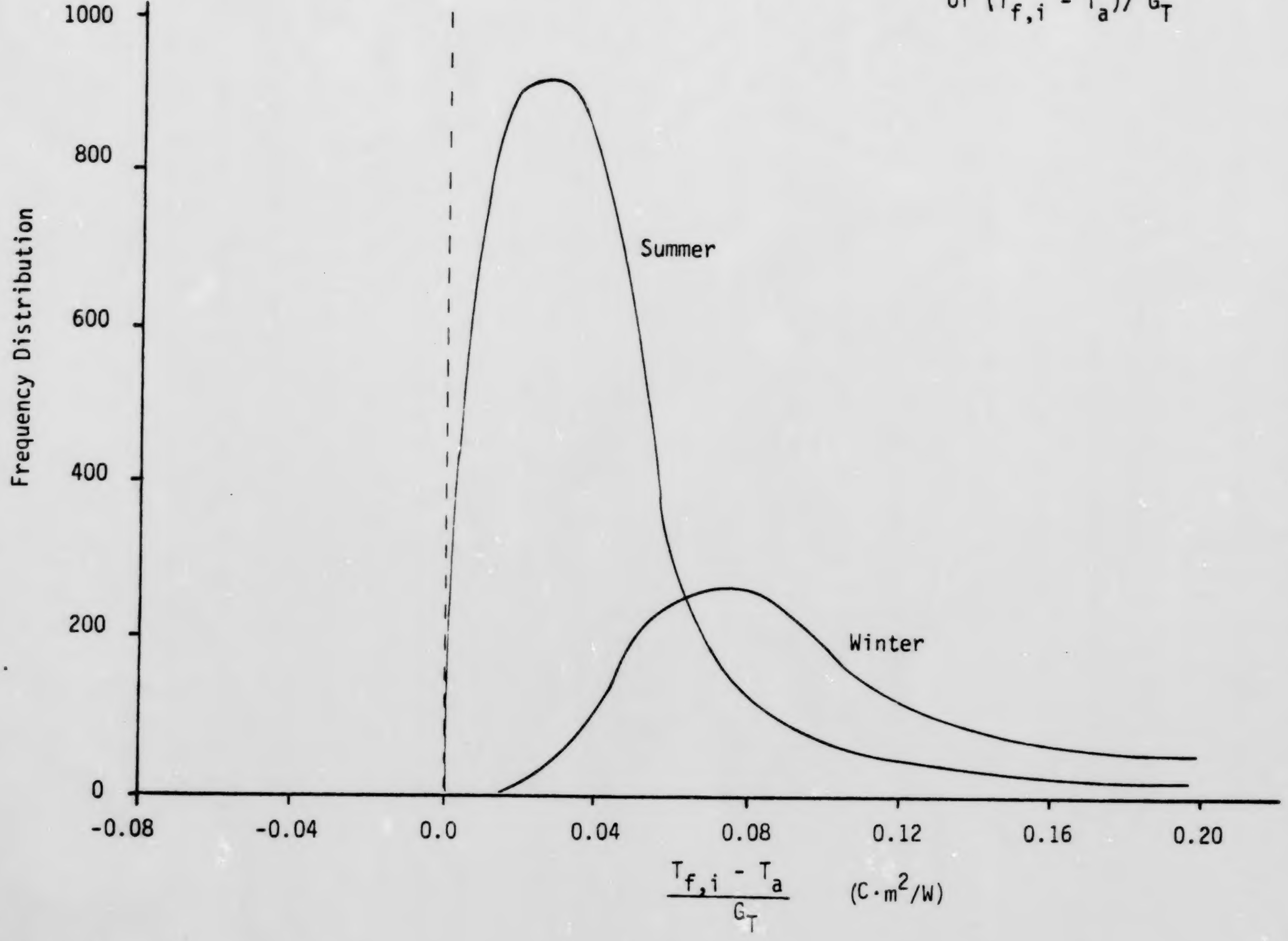
If the same comparison were made at even higher inlet temperatures it would be found that the evacuated tube collector, which has a lower $F_R(\alpha\tau)$ and also a lower $F_R U_L$ than a glazed collector, would have the best range of efficiencies.

This approach to visualising performance can also be applied to the evaluation of collectors for seasonal applications. Figure 3.3.6 shows a frequency distribution of $(T_{f,i} - T_a)/G_T$ split into two separate distributions, for summer and for winter. Obviously, the best collector for one period will not be best for the other. For example, it is likely that for a process water heating application that could be best served by a glazed collector if the load were constant throughout the year, the best collector to serve the same load from May to September might be an unglazed plastic collector, while the best collector to serve that load in the winter might be an evacuated tube collector.

The frequency distribution of collector efficiency provides a useful insight into the performance of collectors in various applications.

FIGURE 3.3.6

Frequency Distribution of $(T_{f,i} - T_a)/ G_T$



However it does not provide a quantitative comparison to the designer. One reason is that in real applications the inlet temperature varies. Another is that the efficiency distribution must be weighted by the insolation distribution to determine the annual performance. Thus the designer must use a simulation program to determine the actual performance of a particular collector for a particular load. The above discussion only provides insight into which collectors may be worth considering for a certain application.

Another issue not considered in this discussion is collector cost. The real issue in selecting a collector is not the amount of energy collected per square foot but the amount of energy collected per dollar invested.

3.5 EFFECTS OF FLOW RATE AND FLUID PROPERTIES ON THERMAL PERFORMANCE OF COLLECTORS

The Canadian Solar Industries Association's Solar Product Data Sheets show the thermal performance curves for solar collectors determined according to the test method described in ASHRAE Standard 93-77. These tests are carried out using the coolant specified by the manufacturer, normally water, and at a single flow rate specified by the manufacturer.

The optimum solar energy system design for a particular application seldom requires the flow rate per collector specified by the manufacturer. Also, in many designs, the coolant is not water. Both these variations change the collector efficiency curve. However, to do this precisely requires a calculation of the flow regime and the heat transfer coefficienct in the collector, and this is time consuming. To assist the design engineer, the Solar Programs Office has produced a Technical Note (No. 2, June, 1982) entitled "Adjustment of Thermal Performance Curves for Variations in Flow Rate and Fluid Characteristics of Solar Collectors". In this note, graphs of $F_a = F_{R\,(collected)} / F_{R\,(test)}$ are plotted against flow rate of water, propylene glycol and ethylene glycol for several PUSH approved collectors. Figure 3.4.1 shows such a graph for a Solarsystems Sol-100 collector.

If, for example, a designer who wished to use propylene glycol circulating at 2 L/min in this collector would read a value of F_a = .97 from Figure 3.4.1, and then calculate

$F_R\,(\alpha\tau) = .97 \times .66 = .64$

and $F_R U_L = .97 \times 4.0 = 3.88$ W/M²°C

Since new collectors are being developed and existing ones are sometimes retested, the Solar Programs Office has produced another Technical Note (No. 3, June 1982) entitled "Calculation of a Flow Rate and Fluid Property Correction Factor for Collector Performance Curves." This note tells the design engineer how to calculate F_a for a collector that is not included in Technical Note 2, or for a flow rate or fluid not covered.

3.6 EFFECT OF WIND SPEED ON COLLECTOR EFFICIENCY

The efficiency of a solar collector is strongly influenced by the wind. A flow of air over the outer cover glass (or over the absorber of an unglazed solar collector) increases the plate-in-air heat transfer coefficient h_{p-a}, and thus increases the loss coefficient U_L. Figure 3.5.1

FIGURE 3.4.1

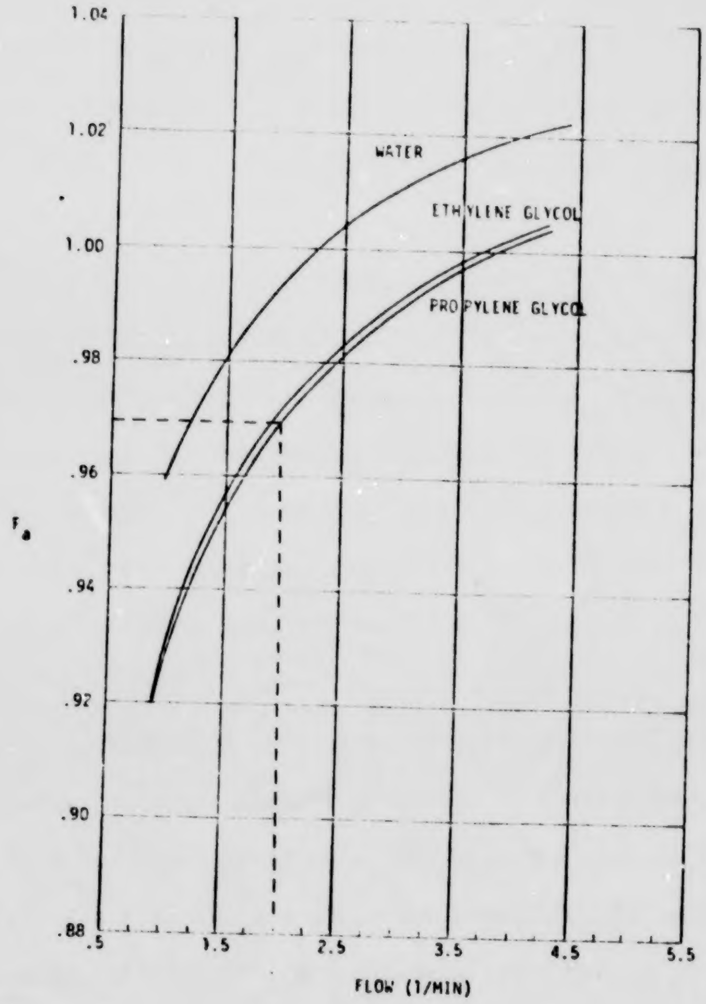
Solarsystems Sol-100 Single-Glazed Liquid Flat Plate

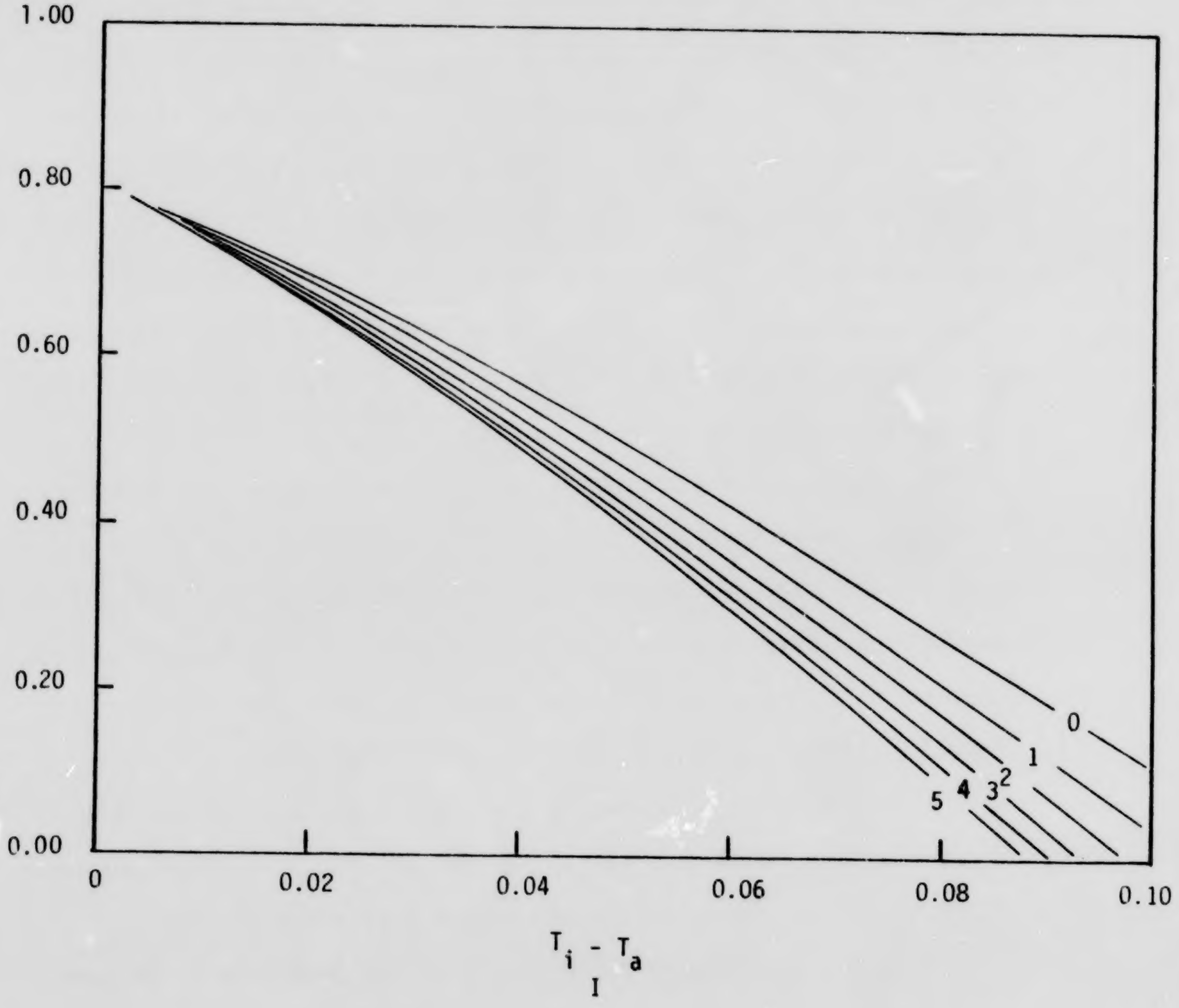
Test Data

Flow rate	2.3 L/min
Fluid	water
Date of test	unknown
$F_r(\tau\alpha)$	0.66
F_rU_L	4.0 $W/m^2\cdot°C$

Source of Information

.CSIA Data Sheet, 5 October 1980





The Effect of Varying Wind Speed on the Efficiency Curve of a Typical Flat Plate Solar Collector

FIGURE 3.5.1

shows the effect of varying wind speed on the collector efficiency curve of a typical glazed flat plate collector.

The ASHRAE 93-77 Standard "Methods of Testing to Determine the Thermal Performance of Solar Collectors" requires that the wind velocity at the collector mid-height be measured and recorded, and that it be less than 4.5 m/s (10 mph). This data is presented in the Canadian Solar Industries Association's Solar Product Data Sheets as a range of wind velocities during the test. A review of this data shows that many tests were run in winds greater than that allowed by ASHRAE 93-77. Also, the range is so great in most cases than it gives little indication of how the data could be corrected for other wind velocities. This presents the designer with a serious problem, since according to Figure 3.5.1, the efficiency of a collector can vary from 0 to 20% even within the range of velocities allowed by ASHRAE 93-77.

The influence of wind velocity on the efficiency of unglazed collectors is even greater than indicated by Figure 3.5.1. ASHRAE Standard 96-80, "Method of Testing to Determine the Thermal Performance of Unglazed Flat-Plate Liquid-Type Solar Collectors" allows wind velocities only up to 1.3 m/s (3.0 m.p.h.) (Designers should beware of comparing 96-80 and 93-77 data, since the standards differ in several respects. The CSA standard on solar collectors, F378 - M1979, covers both types of collectors.)

A new revision of ASHRAE Standard 93 is now in progress. The committee recognizes that the wind velocity variation permitted is perhaps the greatest contributor to variations in the reported results. The new standard will probably require a narrower range of wind velocities during testing, as well as improved measuring techniques. It will also probably specify a velocity of 4±0.5 m/s as the wind velocity for indoor testing.

Until these improvements in testing procedure are implemented, it is difficult for the designer to deal with the problem. Unless the wind velocity at which a test measurement was made is known, it is not possible to correct the results of the wind velocity at which efficiency calculations are required. All that can be done is to recognize that:

1. solar collectors will perform better in regions of low wind then regions of high wind, and
2. variations in wind velocity have induced considerable variation into the reported results of ASHRAE 93-77 collector tests.

The latter point does not suggest that collector efficiency results should be ignored.

Where data on wind velocity during a test is available, the best heat transfer correlation for correcting it is

$$Nu = .86\ R_e^{1/2}\ P_R^{1/3} \ *$$

where $Nu = hD_h/k$ (the Nusselt Number)

$R_e = VD_h\rho/M$ (the Reynolds Number)

$P_r = {}^{M}/\rho\alpha$ (the Prandtl Number)

D_h = 4 x plate area/plate perimeter

h = heat transfer coefficient

k = thermal conductivity

---

* E.M. Sparrow, J.W. Ramsay and E.A. Mass, JGT, 101, 2, 1979, "Effect of Finite Width on Heat Transfer and Fluid Flow About an Inclined Rectangular Plate".

ρ = density

M = viscosity

α = thermal diffusivity

This equation agrees well with theory, and in particular it shows the correct variation of heat transfer with plate size, and the correct variation of heat transfer with the square root of velocity. It was obtained over the Reynolds Number range from 2×10^4 to 9×10^4, but it should be valid over the entire laminar flow range, up to a Reynolds Number of 10^6.

Where collectors are in a turbulent boundary layer, or in the wake of an object, the heat transfer prediction of the above equation should be increased by about 25%.

Solar energy textbooks often recommend an equation of the form $h = a + bV$ for the heat transfer coefficient. This formula ignores the effect of collector plate size, and incorrectly assumes a linear variation with velocity.

Some designers have the mistaken impression that sheltering a surface, such as a solar collector, from the prevailing wind behind some object will decrease the heat loss. This may occur at very low wind velocities; at higher velocities the eddies shed by the object may increase heat transfer. The same is true when the wind is from the back of the collector. At Reynolds numbers over 60,000 the heat transfer on the front surface will be greater than it would have been if the wind had been directly onto the front.*

---

* E.M. Sparrow, ASHRAE Journal, July, 1982, "Wind-Related Heat Transfer Coefficients for Leeward-Facing Solar Collectors".

When the wind is very low, free convection will predominate. A good approximation for the free convective heat transfer coefficient is

$h = 4 W/m^2\ ^{\circ}C$

at a surface-to-air temperature difference of 10°C. This value will vary with the cube root of the temperature difference.

Simplified solar energy system performance analysis methods such as FCHART do not take wind variations into account, although it is not difficult to do so with the hour-by-hour simulation procedures such as TRNSYS. The designer who is working in a climate having unusually high or low winds should make an estimate of the impact of this factor on system performance, using a calculation method with an accuracy level suited to the requirements of the particular job.

3.7 THERMODYNAMIC FACTORS

The standard approach to the analysis of the performance of thermal systems is to introduce the concept of the thermodynamically ideal system (for example, the Rankine cycle thermal power plant). This system operates reversibly, and thus its operation results in no increase of the entropy of the universe. This concept provides a performance standard for the equivalent real thermal system, and provides the designer with insights into how the system should be modified to make its performance closer to the ideal.

It is of little value to apply this concept to solar collectors. An ideal collector would have $F_R(\alpha\tau) = 1$ and $F_RU_L = 0$, but these goals are already obvious to the collector designer, and are also obviously not achievable or even approachable. A collector is a heat transfer device and, like other heat transfer devices such as heat exchangers, it is by definition an entropy producer. The Second Law of Thermodynamics provides little insight into the performance of this kind of equipment.

Second Law analysis is valuable in analysing the performance of the remainder of the system. The ideal solar energy system is one which produces no entropy (outside of that necessarily produced by losses from the collector). Thus, it will have perfect insulation and, more important, it will have no mixing of fluid streams at different temperatures.

To achieve the latter requirement means that there must be no disturbance of the stratification of the storage tank; that the fluid entering the storage tank from the collector must do so at the point where the temperature of the stored fluid equals that of the entering fluid; and that the fluid entering the collector from the storage tank must do so at the point where the make-up fluid has reached the temperature in the bottom of the tank. Thus the ideal solar service water heating system has the configuration shown in Figure 3.6.1, with points A and B as movable injection points.

Of course, such a system cannot be built, but certain approximations to it can be made. For example, where water usage is steady, the two heat exchanger systems shown in Figure 3.6.2 can approximate the injection of water into the collector at the point where the temperatures are equal. This approach has been found to be cost effective in certain circumstances.

The two-tank two-valve system shown in Figure 3.6.3 is an approximation to injection into the storage tank at the point of equal temperatures.

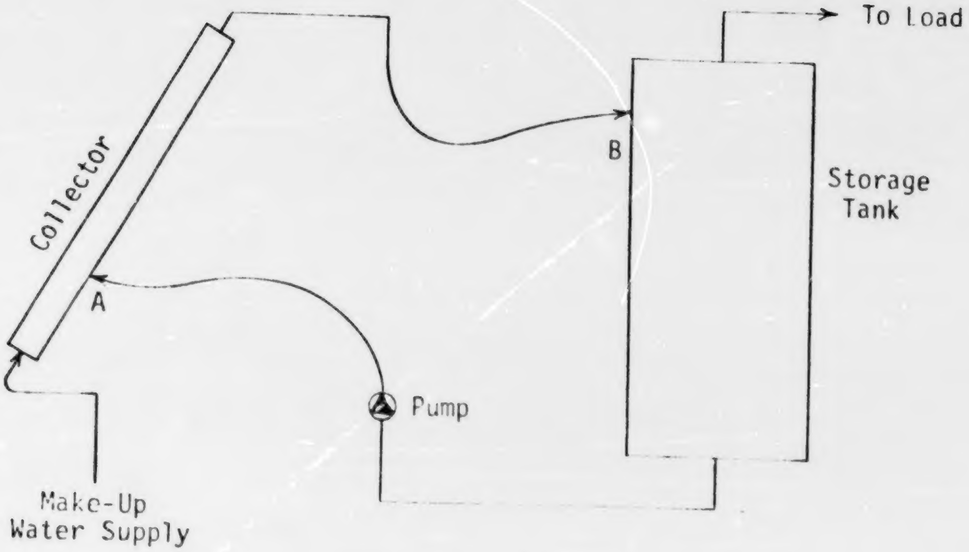

To Load
Collector
B
Storage Tank
A
Pump
Make-Up Water Supply

FIGURE 3.6.1

THERMODYNAMICALLY IDEAL SOLAR

SERVICE WATER SYSTEM

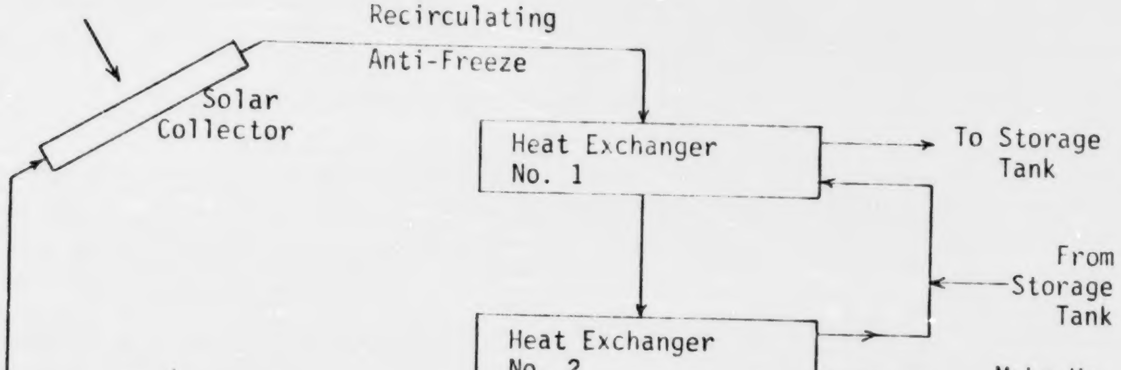

Recirculating
Anti-Freeze
Solar Collector
Heat Exchanger No. 1
To Storage Tank
From Storage Tank
Heat Exchanger No. 2
Make-Up Water

FIGURE 3.6.2

TWO HEAT EXCHANGER SYSTEM

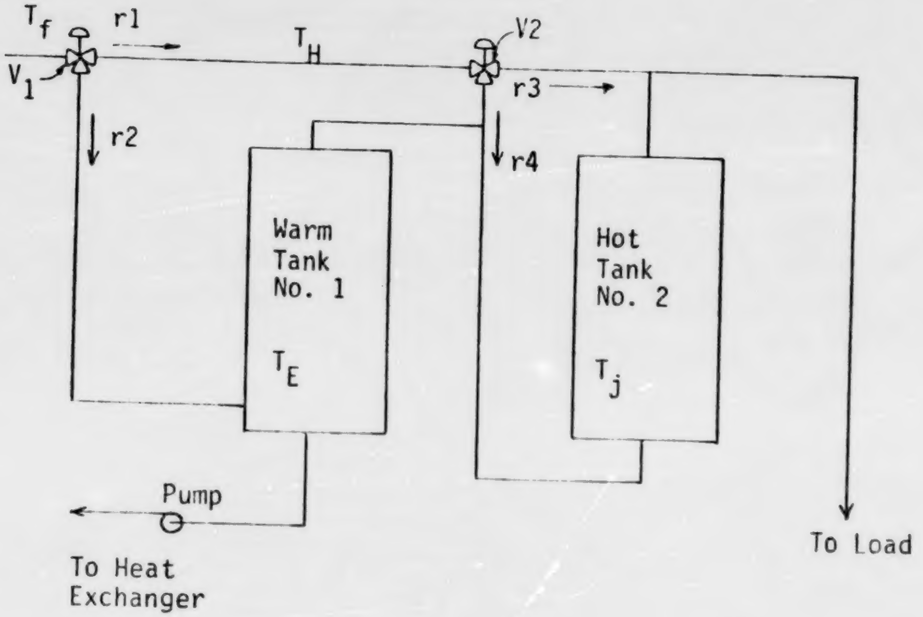

T_f
r1
T_H
V2
V_1
r2
r3
r4
Warm Tank No. 1
T_E
Hot Tank No. 2
T_j
Pump
To Heat Exchanger
To Load

FIGURE 3.6.3

TWO VALVE TWO TANK SYSTEM

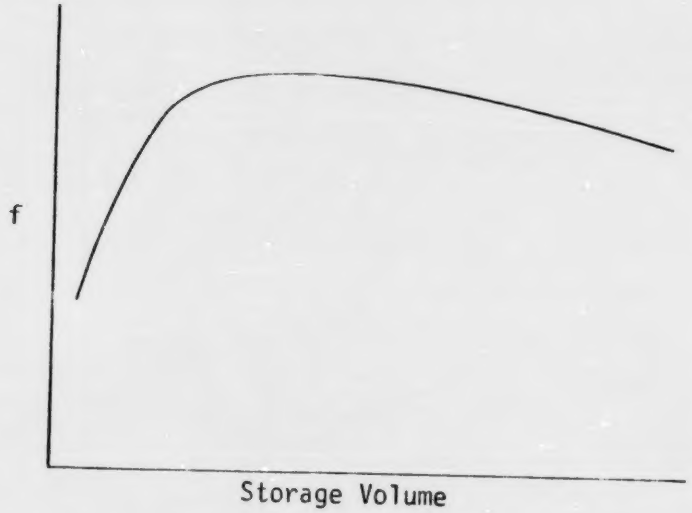


FIGURE 3.6.4

SOLAR FRACTION vs STORAGE VOLUME

FOR A SOLAR DHW SYSTEM

If $T_f < T_E$, then valve V_1 diverts the warm fluid along path r_2, and it recirculates without disturbing the tanks. Any water used will be drawn from the top of Tank 2 and made up at the bottom of Tank 1. If $T_f > T_E$, valve V_1 sends the fluid along path r_1 to valve V_2. If $T_H < T_J$, valve V_2 diverts the flow to path r_4 and the water recirculates through Tank 1, warming it up. Tank 2 is left undisturbed except that any water used will be drawn out of the recirculating loop, into the bottom of Tank 2 and out the top, thus sending the hottest water to the load. If $T_H > T_J$, the valve V_2 diverts water to path r_3 and recirculation is through both tanks, warming them up. Any water used is drawn from the recirculating loop before it enters Tank 2.

This scheme has been found to be cost effective in certain circumstances. Computer analysis indicates that it can more than double the solar fraction compared to the same system without the extra valves and controls. This concept can be extended to larger numbers of tanks and valves for larger systems where it can be shown to be cost effective.

In general, the thermal performance of a system will be enhanced by any factor that results in a lower collector inlet temperature (without wasting heat, of course). One such factor is a low supply water temperature. All other things being equal, a service water system supplied from a well will be a better application than the equivalent system supplied from the mains, in most locations in Canada. Another way of lowering inlet temperature is to combine two different loads in a single solar energy system. For example, a make up air preheater can further cool the collector heat transfer fluid after it has passed through a heat exchanger where it has transferred heat to service water.

Another significant thermal performance factor is optimum storage capacity. In space heating applications an increase in storage capacity will always increase the thermal performance, although not necessarily in a cost-effective way, but in a solar service water heating system there is an optimum tank size above which thermal performance decreases. This optimum size is approximately equal to one day's water usage. The reason there is an optimum is that the only way energy can be withdrawn from storage in a service water heating system is by using water. Thus, if 500 L are to be withdrawn and the required temperature is 50°C, more energy will be gained from a 500 L tank of water at 45°C than from a 5000 L tank at 40°C, even though the 5000 L tank contains far more energy. This reasoning is independent of the fact that a larger tank with the same thickness of insulation will also lose more energy.

Figure 3.6.4 shows a typical variation of solar fraction versus storage volume for a solar service water heating system. Note that it is less dangerous to install too large a tank than too small, if the optimum size is not known exactly.

Choosing the right collector for the job is an important element in the production of a thermally efficient design. Section 3.2 discusses the importance of selecting a collector that performs well in the particular range of the parameter $(T_{f,i} - T_a)/G_T$ which it will experience. An unglazed plastic collector may perform better than an evacuated tube collector for preheating well water in the summer (as well as being cheaper).

A frequently-tried concept in solar energy systems is the variation of flow rate through the collectors. Under certain circumstances this concept could improve system thermal performance if properly applied. In general, however, it is an erroneous idea. The fluid leaves the collectors at higher temperature, but the value of F_R is degraded and the collector efficiency is decreased; thus energy delivered is lowered.

The final topic to be mentioned in this section is controls. Solar heating systems only collect energy when they are operating, so the designer must ensure that a system is running whenever there is energy to be collected. This is not easy. Care must be taken to ensure that the system does not run when it would lose energy, and that it does not cycle on and off at high frequency.

3.8 ECONOMICS

This section covers the materials presented in Chapter 11 of Duffie and Beckman. However, the approach taken is that of "Design of Thermal Systems" by W.F. Stoecker, McGraw-Hill, 1971.

The basic problem of solar energy economics, like that of the design of any other system where capital investments made now can reduce future running costs, is to compare expenditures and incomes made at different times. This is particularly true in an inflationary economy.

To find the present value of a payment made n years in the future, the amount of the payment is multiplied by the present worth factor,

$$\text{P.W.F.} = \frac{1}{(1+y)^n} \qquad \text{(Eqn. 3.7.1)}$$

where y is the interest rate. Similarly, dividing an investment made today by the PWF will predict its value n years in the future.

If a series of annual payments are to be made over the next n years, the present value can be found by multiplying the annual payment by the series present worth factor,

$$\text{SPWF} = \frac{1 - (1+y)^{-n}}{y} \qquad \text{(Eqn. 3.7.2)}$$

Similarly, dividing a debt incurred today by the SPWF will determine the annual payments necessary to pay it off in n years.

If it is required to calculate the present value of a fuel saving worth $1.00 this year that will continue for n years, inflation as well as interest must be taken into account. This can be done by defining y as the effective interest rate

$$y = \frac{i - x}{1 + x} \qquad \text{(Eqn. 3.7.3)}$$

where i is the nominal interest rate and x is the inflation rate for the commodity in question (in this case, fuel). This expression, and the equations for PWF and SPWF are valid for $n>i$ and for $n<i$, but if $y=0$ the SPWF equation must be replaced by SPWF=n.

Example

A solar energy system costing $100,000 is projected to last 20 years and require major maintenance after 10 years. This major maintenance would cost $10,000 today and its cost is escalating at 6%. The annual maintenance cost is $500, escalating at 6%. The system saves $6870 per year of oil, escalating at 8%. The discount rate is 10%. Is this a good investment.

Present values:

1. Major maintenance

$$P.V._{mm} = 10000 \times PWF_{mm}$$

$$Y_{mm} = \frac{i - x}{I + x} = \frac{.1 - .06}{1.06} = .0377$$

$$PWF_{mm} = \frac{1}{(1+y)^n} = \frac{1}{(1.0377)^{10}} = .6907$$

$$\therefore P.V._{mm} = 10,000 \times .6907 = \$6907.$$

2. Minor maintenance

$$P.V._{a.m.} = \$500 \times SPWF$$

$$y_{a.m.} = .0377$$

$$\therefore SPWF_{a.m.} = \frac{1 - (1+y)^{-n}}{y}$$

$$= \frac{1 - (1.0377)^{-20}}{.0377}$$

$$= 13.8714$$

$$\therefore P.V._{a.m.} = \$500 \times 13.8714$$

$$= \$6936$$

$$\therefore P.V._{costs} = 100,000 + 6907 + 6936$$

$$= \$113,843$$

3. Fuel Savings

$$P.V._{f.s.} = 6870 \times SPWF$$

$$Y_{f.s.} = \frac{.1 - .08}{1.08} = .0185$$

$$\therefore SPWF_{f.s.} = \frac{1 - (1.0185)^{-20}}{.0185}$$

$$= 16.5876$$

$$\therefore \text{P.V.}_{\text{f.s.}} = \$6870 \times 16.5876$$

$$= \$113,957$$

$\therefore \text{P.V.}_{\text{f.s.}} > \text{P.V.}_{\text{costs}}$ and the system is a good investment.

Assignment

Oil costs 12¢/L (39000 kJ/L) and is escalating at 14%/year. A solar service water heating system will produce 1000 MJ/day of heat. It costs $100/year to maintain (escalating at 8%). The salvage value of the solar energy system will be $1000 in 20 years. (In making this estimate, inflation has been taken into account, so this figure is in year 2001 dollars, not year 1981 dollars.) The efficiency of the backup oil burning system is 60%. The interest rate is 11%. How much can you pay for the solar energy system?

Figure 3.7.1 shows a typical economic optimization plot. Summing the present value of future fuel expenditures and the system capital cost produces a curve which has a minimum at a particular collector area. The value is below the present value of fuel expenditures without a solar energy system, so the collector area at the minimum is obviously the correct amount to install.

Another possible economic optimization curve is shown in Figure 3.7.2. In this case, even the optimum solar energy system size results in a greater present value of total costs than no solar energy system. If there are non-economic motives for building a solar energy system, the optimum point still provides guidance as to the best system size. However, when the (positive) slope of the solar energy system capital cost curve is always greater than (negative) slope of the present value of fuel costs curve, there will be no optimum. In this case the designer who has reasons other than economic ones for building a solar system will also have to use a non-economic approach to determining system size.

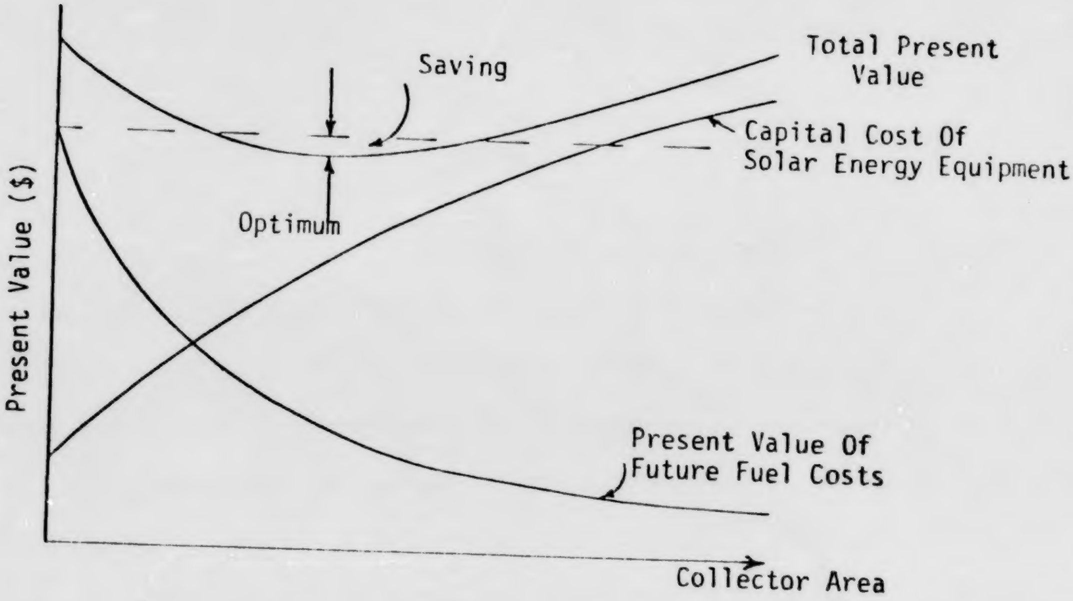


FIGURE 3.7.1

ECONOMIC OPTIMIZATION

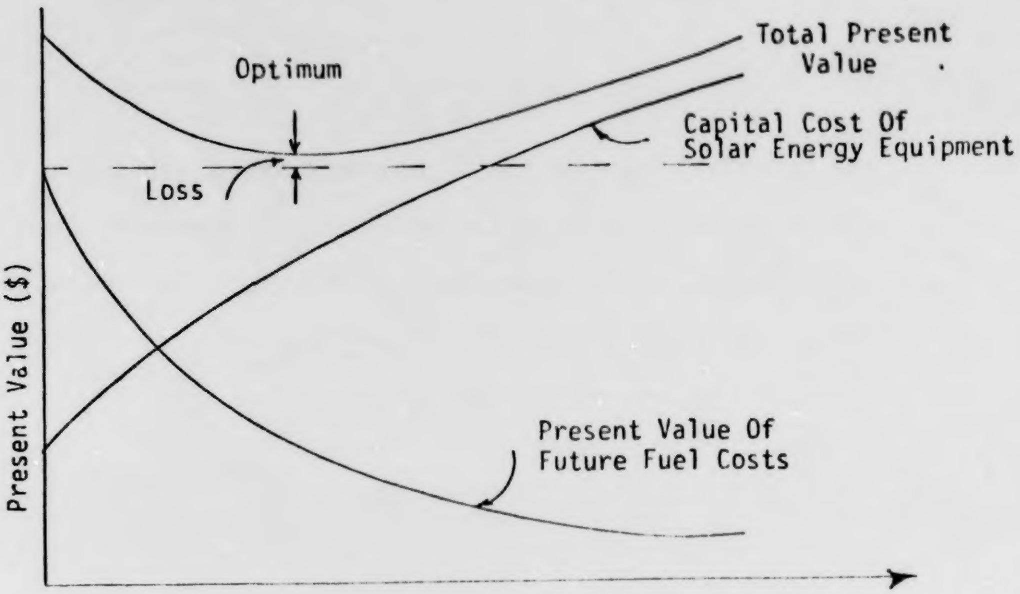


FIGURE 3.7.2

ECONOMIC OPTIMIZATION

3.9 SYNTHESIZING THERMAL AND ECONOMIC FACTORS INTO A SYSTEM DESIGN

A designer who wished to develop an optimized solar energy system would require a fairly complex computer program to do so. First, a precise thermal model is needed to distinguish the effects of the variables to be optimized (for example, the effect of an extra temperature controlled diverting valve in a multi-tank storage system). This program would then be built into an optimization program using one of the several optimization techniques available (See Stoecker, Page 108), and the program supplied with the costs of all the possible components. This approach is widely used in the design of thermal storage. For example, a refrigeration equipment manufacturer will use this approach in selecting the components of a heat pump.

The problem with applying this approach to solar energy system design is that design costs are part of the optimization equation. Larger solar energy systems are unique designs and only a limited effort can be spent on optimizing them before the design costs begin to outweigh the system cost savings. Even solar domestic water heating systems are not sold in large enough volumes to justify major expenditures on optimization.

Lacking the opportunity to do detailed analysis of cost-effectiveness, the solar energy system designer must rely on judgment, experience, and rules-of-thumb to produce the most economic systems possible. Many factors that make systems more economic are related to the site, so that good design becomes more a question of recognizing good applications and good sites than one of fitting systems to pre-determined sites. Among the important factors to look for in a site are:

a) an existing roof or other structure with the right slope and orientation, to save the cost of a support system;

b) sufficient space for the mechanical equipment, close to the collector array, and with easy accessibility;

c) a load that is close to the room where the solar energy system will be installed;

d) no shading from adjacent structures, and a low probability of future construction;

e) a site and a load with projected lifetimes greater than the projected solar energy system lifetime;

f) a low probability of vandalism;

g) the availability of maintenance staff; and

h) sub-metering on the load to be met, so that system sizing is easier and future monitoring of performance will be cheaper.

Among the important factors to look for in the load itself are:

a) a low temperature load (note the preheating boiler feed water can be a low temperature load. The required temperature is less important then the inlet temperature).

b) a large load, which permits economies of scale and also permits keeping the mean system temperature low,

c) a good time variation of the load. (The best time variation would be one that follows the availability of sunlight both through the day and through the year. The worst would be one that was concentrated in the winter and at night, such as a space heating load) and

d) a load that is being met by a high priced fuel and for which no cheaper alternatives are readily available.

Given a good site and a good load, it remains to the designer to apply the principles of good thermal performance outlined in Section 3.6 above, and to design the cheapest possible system consistent with meeting the design objectives outlined in Section 3.2. There are two final principles which the designer should keep in mind. One is that a high solar fraction is not related to cost effectiveness. It may well be worthwhile, for example, to abandon the winter portion of a service water heating load to the electrical backup system, and avoid the cost of a separate heat transfer loop with freeze protection. The second point is that the PUSH Program has as its objective the development of a solar equipment manufacturing industry, not a plumbing hardware industry, so that trade-offs between equipment investments should be slanted towards the collectors rather than the associated equipment.

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CHAPTER 4

SIMULATION TECHNIQUES

Several computer programs are available to assist in the analysis and design of solar energy systems. Two of the most widely used programs are FCHART and TRNSYS. FCHART is based on a correlation of the results of hour by hour annual TRNSYS simulations. Other frequently used programs are WATSUN, DOE-2, BLAST, ENERPUB, VENTAIR and ASTEPI.

4.1 THE TRNSYS COMPUTER PROGRAM

TRNSYS is a transient system simulation program, which was developed at the University of Wisconsin Solar Energy Laboratory. TRNSYS was then used to develop the FCHART program, which is not as versatile as TRNSYS but is much easier to learn and use. FCHART is used for the analysis and design of active solar and heat pump heating systems. It is a self-explaining interactive program which allows the user to quickly describe the system and which provides a straightforward table of results.

By comparison, TRNSYS requires some knowledge of computer programming. TRNSYS is capable of simulating almost any solar thermal energy system. A system is described by a set of components which are interconnected. The programmer specifies which components are included in the system and the manner in which they are interconnected. Most systems described in this manner can be simulated using components found in the TRNSYS library. However, other components may also be modelled by writing Fortran subroutines to simulate the desired components. The TRNSYS library includes the following component models: data reader; solar radiation data processor; shading algorithm; flat plate collector; CPC collector; mass wall; direct gain window; pipe and duct; pump/fan; flow diverter/mixing valve/tee piece; controller with hysteris; three-stage thermostat; relief valve; heat exchanger; storage tank; rock bed; heat pump; absorption air conditioner; auxilliary heater; liquid collector-storage subsystem; air collector-storage subsystem; domestic hot water subsystem; energy (degree-hour) space heating or cooling load; cyclic time-dependent function generator; algebraic operations unit; quantity integrator; printer; plotter; time and frequency distribution plotters; simulation summarizer and wall; roof-and-attic, room and basement-transfer function models.

The TRNSYS program also requires that the programmer select the format and contents of the output, while FCHART is programmed to provide a monthly breakdown of annual system performances. However, TRNSYS is capable of simulating the system for any desired period using any chosen timestep. For example, the system can be simulated hourly for a day, daily for a month, weekly for a year, etc.

4.1.1 Advantages of TRNSYS

1. The programmer is able to select the content and the format of the TRNSYS simulation output.
2. The programmer is able to select the timestep and the length of the simulation used by TRNSYS.
3. TRNSYS's modular system provides a tool for easy simulation. The entire problem of system simulation reduces to a problem of identifying all components and formulating a general mathematical description of each.

4. In TRNSYS's modular system, many components are common to different systems and can thus be used in these systems with little or no modification.

5. Transient systems can be simulated by TRNSYS in great detail.

4.1.2 Disadvantages of TRNSYS

1. Learing the TRNSYS program is a long process, and may require a learning period five to ten times greater than FCHART.

2. Running the TRNSYS program is very expensive compared to other available programs which may be used, such as FCHART. For example, comparable annual simulations of a solar DHW system typically cost over $30 using TRNSYS and less than $1.00 using FCHART. Research of the University of Wisconsin Solar Energy Laboratory indicates a very close correlation between FCHART and TRNSYS - usually within a few percent.

3. TRNSYS component models or subroutines not in the library are difficult to program, and require relatively large amounts of computer time. The compact subsystems found in the TRNSYS library are most efficient and therefore use less computer time. The solar DHW system referred to above was based on the TRNSYS solar DHW subsystem. The system was rerun using separate storage tank and auxiliary heater subroutines, and the cost was just over $50.

4.2 THE FCHART COMPUTER PROGRAM

This computer simulation program is the correlation of the results of many hundreds of thermal performance simulations using the TRNSYS program. These simulations were performed for a range of parameters which are found in practical system designs. The most important design parameter is collector area. Other variable parameters include collector type, storage

TABLE 4.2.1

RANGES OF DESIGN PARAMETERS USED IN DEVELOPING FCHART

$0.6 \leq (\tau\alpha)_n \leq 0.9$

$5 \leq F'_R A \leq 120\ \text{m}^2$

$2.1 \leq U_L \leq 8.3\ \text{W}/\text{m}^2\text{C}$

$30 \leq \beta \leq 90\ \text{deg.}$

$83 \leq (UA)_h \leq 667\ \text{W}/\text{m}^2\text{C}$

capacity, fluid flow rates, and load and collector heat exchanger sizes. Ranges of the design parameters used in developing FCHART are shown in Table 4.2.1.

The FCHART simulation program incorporates three basic system configurations - liquid and air systems for space and hot water heating and systems for service water heating only. FCHART calculates the monthly system performance from long term averages of monthly meteorological data and the input parameters.

FCHART is based on the following assumptions:

1. All liquid storage tanks are assumed to be fully mixed.
2. The hot water demand in a DHW system is assumed to be the daily mass flow profile developed by a Rand Corporation survey normalized and applied for a user-specified daily mass demand (see Figure 4.2.1).
3. The solar collectors are assumed to have a straight line efficiency curve.
4. The solar radiation is considered to be symmetrical about solar noon.
5. The systems are well designed and well-built, the flow distribution is uniform, the flow rate corresponds to the collector performance data used and the systems are similar to the computer models.

4.2.1 FCHART Utilizability

Utilizability is defined as the fraction of the incident solar radiation that can be converted to useful heat by a collector with $F_R \tau\alpha = 0.9$ and a fixed inlet to ambient temperature difference. FCHART uses the monthly average utilizability as a parameter in the design of liquid systems for space heating and combined space and water heating.

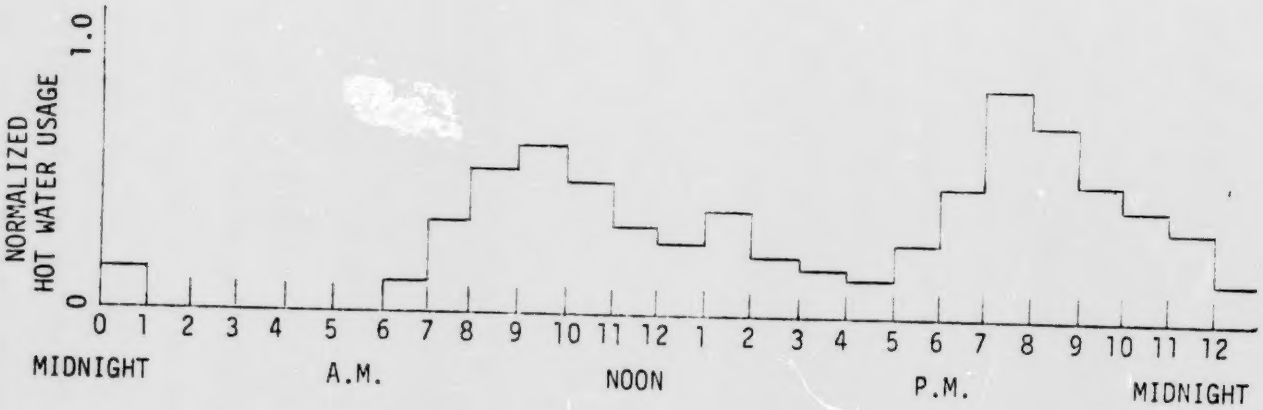


FIGURE 4.2.1 Daily Mass Flowrate Profile

The following two correlations for utilizability are used in FCHART. For flat plate collectors, utilizability is defined as the fraction of radiation, above a fixed critical level, which is incident upon a collector surface. This critical level is found by setting the collector useful output (Q_u) equal to zero in the following equation,

$Q_u = A_c F_R [I_t (\tau\alpha) - U_L (T_i - T_a)]$. For concentrating collectors, utilizability is defined as the fraction of radiation, above the fixed critical level, which is incident upon a collector surface over the period of time that the collector operates.

4.2.2 FCHART Output Analysis

The FCHART computer simulation output consists of an evaluation of the system thermal performance, a summary of the purchased energy and, optionally, a life cycle analysis. These results depend on the type of system chosen, the options selected and the parameter values which were used. Once the proposed system has been designed and the input parameters have been chosen, the system may be simulated.

The monthly system performance is evaluated using the equipment characteristics which were input and the long-term average of recorded weather data. FCHART is a very good tool for the long-term prediction of solar energy system performance.

The economic analysis of a design may be hand calculated, or obtained from the FCHART output. Use of the FCHART economic analysis will depend on the needs of the user. If a simple payback period is sufficient, the FCHART economic analysis need not be simulated. However, if a more in depth analysis is required, including corrections for depreciation, interest and inflation, the FCHART simulation should be used.

The FCHART life cycle cost analysis output has three levels of detail which may be selected according to the user's requirements. FCHART also has the capability to compare the life cycle costs of two systems. This comparison may be obtained in two degrees of detail. A detailed explanation of the economic analysis may be found in the user's manual.

4.3 THE WATSUN COMPUTER PROGRAM

WATSUN is a series of generalized computer programs developed at the University of Waterloo as a design aid for the study of solar-assisted domestic hot water and space heating systems. WATSUN uses many of the algorithms used in the TRNSYS and FCHART computer programs. Like FCHART, WATSUN uses total system configurations rather than the TRNSYS-type component models. WATSUN uses actual weather data to simulate the hour-by-hour operation of the system. Like TRNSYS, the starting day, length of simulation period and output timestep are specified by the WATSUN user. However, certain assumptions in WATSUN can make the validity of short-term simulations (less than a few weeks) questionable. In addition, the economic analysis uses a one year base to project the economic performance of the system over its lifetime. Thus, the economic analysis is produced only for simulations that are exactly one year in length.

WATSUN requires slightly more detailed input than FCHART. WATSUN has the capability to alter the hourly and daily domestic hot water load profiles. WATSUN also has the capacity to simulate identical systems with incremental changes in a variable. WATSUN will simulate the system operation for a given collector area and storage-volume to collector-area ratio. If specified, the collector area is then increased (the storage volume to collector-area ratio remains constant) and the simulation is repeated. This cycle continues until the maximum collector area is simulated. The collector area is then reset to its initial value and the cycle is performed for incremental changes in the storage-volume to collector-area ratio. The cycle ends when the combination of the maximum area and the maximum volume to area ratio has been simulated.

4.4 THE ENERPUB COMPUTER PROGRAM

The ENERPUB computer program, developed by Enermodal Engineering Limited for Public Works Canada, can be used to simulate the performance of liquid-based seasonal or short-term storage solar space heating systems with or without a heat pump. The program can also simulate the performance of a solar heating system used to heat oil in oil storage tank. ENERPUB is not suitable for simulation of solar domestic hot water systems specifically because of the restrictive heat exchanger models.

The ENERPUB program has many advantages over other solar simulation programs. Because performance calculations are made for each simulated hour, the program results are potentially more accurate than those of programs using a monthly calculation procedure, such as FCHART. In addition, the ENERPUB program can simulate a wider range of parameters than programs restricted by monthly average correlation equations. Several of the models in the ENERPUB were adapted from WATSUN models (WATSUN is an hour-by-hour computer program developed by the University of Waterloo). ENERPUB has several enhancements over the WATSUN space heating programs:

1) Stratified tank model for short term or seasonal storage.
2) Four possible heat pump locations.
3) Inclusion of heat loss from the storage to the building, outdoors or ground.
4) Inclusion of heat loss from collector and storage piping to the building, outdoors or ground.

The ENERPUB computer program gives detailed and accurate results for liquid-based seasonal or short-term storage provided that reasonable values for all system parameters are input. All users should read and understand this manual before using the program.

4.5 THE VENTAIR-2 COMPUTER PROGRAM

The VENTAIR-2 computer program was developed by Enermodel Engineering Limited for Public Works to simulate the performance of air-based solar collectors used in the preheating of industrial process or building ventilation air. The program can accept data of all PUSH approved air-based collectors (Amherst 200, Solartech Solair, and Watershed A100) and any other parallel plate air-based collector.

VENTAIR-2 is the only computer program capable of simulating solar preheating of process or ventilation air. Performance calculations use a one-hour time step to ensure maximum accuracy. An important restriction, however, on the use of the program is that the air demand must be constant over the hour. A system not applicable to the VENTAIR-2 program, for example, is a bathroom ventilation system operating by a light switch or short duration timer.

The VENTAIR-2 computer program will provide an accurate estimate of thermal performance of solar preheated process or ventilation air systems provided that the user supplies system parameters within the program limitations.

In many buildings a high rate of change of ventilation air is necessary for the health of the occupants or proper operating of equipment. Such buildings include hospitals, vehicle maintenance garages, indoor swimming pools and commercial kitchens. The volumes of air involved are often quite large (6 air changes/hour) and correspondingly large quantities of energy are required to raise the air temperature to the building temperature. This air is normally heated by steam coils or electric resistance heaters.

To reduce energy consumption, heat recovery equipment can be used to transfer some of the heat from the exhaust air to the fresh air. Heat recovery equipment, however, has several limitations. This equipment suffers from frost build-up in sub-freezing conditions, fouling from kitchen grease, and requires modifications if the exhaust air contains toxic chemicals. The installation of this equipment can be expensive in retrofit situations if return air duct work must be added.

Use of air-based solar collectors to preheat ventilation air has been proposed as a parallel application of a viable alternative to heat recovery equipment in many applications. In making a decision to use this type of system, it is necessary to estimate the yearly thermal performance. The

VENTAIR-2 computer program was designed specifically to assist the designer in evaluating the system performance of solar preheated ventilation air systems.

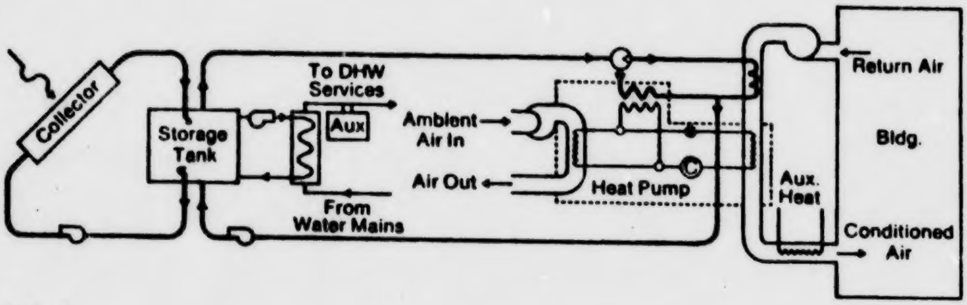
Many industrial processes require large volumes of air for drying and/or heating. Such processes include paint drying, food drying, combustion air preheating and sterilization/drying of bottles and cans. The VENTAIR-2 computer program can also be used to simulate solar preheating of these processes provided that the air flow is constant over each hour. If the flow rate is dependent on the relative humidity and/or ambient temperature the VENTAIR-2 computer program is not suitable.

4.6 ASTEPI

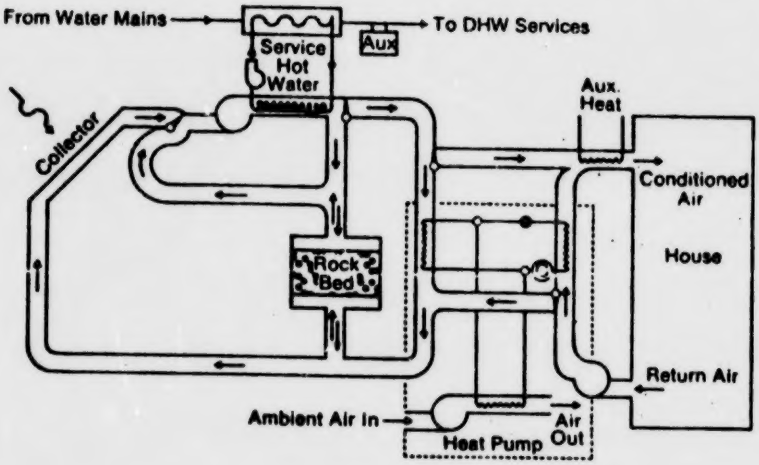
ATTEPI (Annual Storage solar Energy Program) is a computer program written in PL/1, at the University of Toronto designed to simulate the hourly performance of solar heating systems for space and service water heating. Although emphasis is directed to seasonal storage systems, options are included to permit the simulation of short term storage systems as well as a combined seasonal storage and short term storage system.

ASTEPI is designed to utilize the newly expanded solar radiation data base of the Canadian Atmospheric Environmental Service (AES).

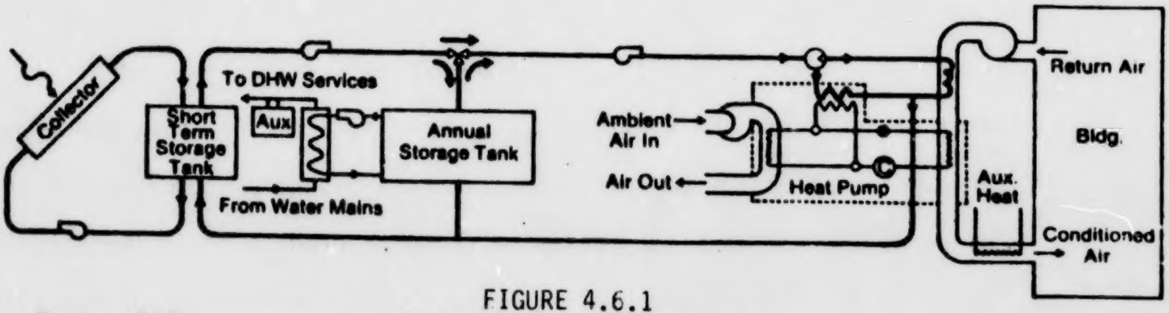
The simulation proceeds in an hourly stepwise fashion, assuming in any given hour that the weather inputs (outside dry bulb temperature and solar irradiance) and other system parameters (i.e. hot water storage temperature, building energy demand) are constant. This reduces the time dependent differential equations associated with the thermal performance of the


Collector
Storage Tank
To DHW Services
Aux
From Water Mains
Ambient Air In
Air Out
Heat Pump
Aux. Heat
Return Air
Bldg.
Conditioned Air

System I


From Water Mains
Service Hot Water
Aux
To DHW Services
Collector
Aux. Heat
Conditioned Air
Rock Bed
House
Return Air
Ambient Air In
Heat Pump
Air Out

System II


Collector
Short Term Storage Tank
To DHW Services
Aux
From Water Mains
Annual Storage Tank
Ambient Air In
Air Out
Heat Pump
Aux. Heat
Return Air
Bldg.
Conditioned Air

System III

FIGURE 4.6.1
ASTEPI BASIC SYSTEM TYPES

system to the simpler algebraic equations which can be solved directly. Although the large thermal storage tanks associated with annual storage undergo little temperature change on a daily basis, let alone hourly, there is advantage in calculating hourly instead of daily. This is due to an increased accuracy in the collected solar energy calculation. Experimental collector performance data (slope, intercept of efficiency curve) based on the collector inlet fluid temperature are used to model the collector performance.

Storage heat loss data, based on detailed conductive soil heat transfer calculations for buried right bylindrical tanks, are used as input to a lumped parameter effective thermal resistance (ETR) model for estimating tank heat losses.

ASTEPI is designed to simulate a number of cases in a single run, allowing the operator to examine the effects of changing one or more parameters.

4.7 THE BLAST COMPUTER PROGRAM

The Building Loads Analysis and System Thermodyamics (BLAST) program was developed by the U.S. Army Construction Engineering Research Laboratory under the sponsorship of the Air Force Engineering and Services Center, Tyndall Air Force Base, Fl. and the Department of the Army, Office of the Chief of Engineers, Washington, D.C. BLAST is a set of subprograms used for predicting energy consumption and energy system performance and cost in buildings. BLAST is a very detailed computer program that simulates hour by hour space heating and cooling loads, hourly performance of fan systems, hourly conventional heating and cooling plants, total energy plants and solar energy systems, and life cycle costs.

BLAST may be used for both new and existing buildings. During the preliminary design phase of a new building it may be used to perform cost-benefit analyses of design options such as site location and building orientation, size, shape and fenestration. It can also be used to determine the best operating strategies and to size the HVAC equipment. In existing buildings BLAST can be used to determine changes in operating strategies.

BLAST is able to simulate solar collectors as well as such passive solar energy system details as Trombe Walls and movable insulation. While BLAST is not normally used to simulate the performance of a solar energy system, it is used to compare the performance of a solar energy system to that of a conventional HVAC system. Details of the BLAST program may be found in the user's manual.

Public Works Canada

Solar Programs Office

Travaux publics Canada

Bureau des programmes solaires

Public Works Canada
Solar Programs Office

SOLAR DESIGN COURSE MANUAL

Part III: ADVANCED (Practical Considerations)

June 1983

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   APPENDIX A

1. EVALUATING OPPORTUNITIES FOR SOLAR ENERGY SYSTEMS

Much of the success of a solar heating system rests with the initial feasibility study. A number of items must be addressed during the feasibility study stage of a solar system project. These include answering the following questions:

- Is the load being considered a good application for a solar heating system?

- Can the solar heating system be easily and cost effectively integrated with existing energy systems?

- Is the required space available for the collector array, necessary heat exchangers, thermal storage and other system components?

- If the collector array is to be roof mounted, is the existing roof structure adequate to carry the collector array without significant modifications?

- Can piping be installed in such a way that required slopes are available, particularly in the case of drainback systems?

- After the system has been installed, will adequately qualified maintenance staff be available to operate and repair the system?

A NO answer to any of these questions may indicate that the load being assessed for application of a solar heating system is indeed not a good one.

1.A LOAD EVALUATION

Today, by far, the largest effort in the active solar heating area is in the production of domestic or process hot water. The systems range in size from extremely large engineered systems such as the Mohawk Hospital Services System constructed under the Ontario Demonstration Programme consisting of over two thousand square metres of collector area, to small package systems consisting of approximately four square metres of collector area.

One of the areas most often neglected, in the feasibility study phase of a solar project, is accurate assessment of the total load and particularly load profiles. Many systems have been carefully designed based on totally erroneous load profile estimates. Often, because details of load profiles were not necessary to the operation of the existing energy systems, the system owner or operator had very little knowledge of the details of these load profiles. Experience has shown, in a very expensive manner, that if good load profile data is not available it must be developed before a solar heating design is proceeded with.

It is also important to consider at this point whether the load profile in an existing plant can be modified to more closely match available solar energy without adversely affecting the plant's operation. These changes, if acceptable, must be determined at the outset and the appropriate solar system design consideration made.

Other changes anticipated in the plant's operation must also be assessed at the time of the feasibility study. As an example, addition of heat recovery equipment in an existing plant will seriously affect the operation of a solar heating system. In some cases it will preclude the use of solar heating and in others radically change the required design.

The following criteria should be used when evaluating a load:

- The load should be large

- The load should be continuous or peaking when solar energy is available

- The load should be centrally located

- The load should utilize low temperature energy

One must also consider how the load would be affected if energy conservation procedures were applied.

In order to match the solar system size to the load required, we should compare energy load requirements on a daily basis and on a seasonal basis. When matching load to the available solar energy certain parameters can be manipulated:

- Time at which the load is used

- Amount of collector area

- Use of storage

The time of day in which the load is to be used is one of the most important to manipulate. If such an opportunity is available it would possibly limit or eliminate storage and make the solar system much more cost effective.

F-Charts or Watsun programs can help decide whether extra energy is to be stored or not.

For space heating requirements, peak solar loads occur at night and solar radiation occurs during the day. In this case, thermal storage is definitely required.

Here are some constraints to system sizing, aside from the load:

- With retrofit systems mounting area available is often limited,

- Methods and costs of integrating with existing mechanical systems can limit both feasibility and sizing of systems.

1.B BUILDING INTERFACE

1.B.1 Required Space for Collector Arrays, Heat Exchangers, Storage, etc.

Solar collectors may be roof mounted, ground mounted or in some cases wall mounted. In all cases the basic criteria are the same.

- Adequate support must be available

- Shading must be minimal

- Protection from vandalism must be adequate

- Presence of potential dirt sources must be minimal

Structural considerations for roof mounted arrays will be dealt with in detail in the next section, however, even for ground mounted array foundations requirements must be carefully considered at the outset.

With ground mounted arrays potential frost heaving must be considered because of its effect on pipe slopes. Particularly for drainback type systems where pipe sloping is critical, minor frost heaving can have disasterous effects.

Shading from large structures, such as adjacent buildings or large trees, are easy to assess and very obvious, however, small items such as roof vents and chimneys can cause shading problems on one or two collectors and lead to system operational problems.

It is also important to consider future potential obstructions to sunlight such as new buildings where a site is evaluated.

Vandalism has seldom been a problem with solar collector arrays but situations can exist where significant damage can occur.

Vandalism problems can usually be minimized by some careful thought when locating the array.

- Don't place collectors in school yards or other locations where children play;

- In public areas, place collectors well back from the edge of roofs;

- Ensure that roof access is made difficult for the unauthorized;

Often overlooked, where solar systems are designed for industrial applications, are sources of dirt which can cover a collector array and significantly decrease its performance.

The most common sources of dirt are combustion vents, and industrial dryer vents.

Although most surface contamination on collector glazing will be washed off by rainfall, heavy buildups of soots or lints will not, and hence, collectors will required frequent and time consuming washing.

Whether a solar system be large or small, adequate space for the location of components is an absolute necessity. Numerous systems have been installed with little consideration for future maintenance of components.

Some examples of bad practices we have seen include systems where components were arranged so that a major effort is required to access items needing relatively frequent attention, such as pumps and heating coils. Package solar heating systems have been installed where it is necessary to remove most of the solar heating system to access the heating element in a domestic electric heating tank. Even when adequate space is available logical placement of components for ease of maintenance must be considered. As an example, in one system the fill valve, which requires relatively frequent attention, was installed at ceiling level.

Generally, choosing a location for the solar storage tank is one of the easier decisions. It is usually placed in the existing mechanical room near major services such as electricity, gas and, of course, water mains. Some rules of thumb do apply and should be adhered to when locating storage tanks:

1. Locate the storage tank as close as reasonably possible to where the present water heater is located. This will reduce the amount of labour and material needed to connect the solar system to the hot water plumbing and the present water heater if it is to remain as the auxiliary heater. This will also reduce heat loss from long pipe runs.

2. The solar storage tank must be located so it is accessible and does not block or hinder access to other components. The storage should not impede passage through hallways or doorways, or be located in such a place as to create a safety hazard.

3. Many solar systems require connections to floor drains in case of an

overflow. The location and accessibility of these drains should be determined during the site evaluation.

Space must also be available for pumps, heat exchangers and for a variety of fluid regulating devices, some of which are large such as expansion tanks.

1.B.2 Structural Considerations For Roof Mounted Arrays

When determining the location for a roof mounted solar collector array, a detailed assessment of the existing structure is required. Far too many solar heating systems have experienced major cost overruns due to difficulties with the structural requirements. While solar collector arrays are becoming far less complex with new innovative designs for collector manifolding and mounting, a sound structural foundation is still a primary concern.

Small Residential Systems

Roofs have little additional roof loading capacity so great care must be taken with choosing how to mount the array. Roofs on modern buildings are generally designed to meet the minimal requirement of applicable building codes. Loads imposed by solar arrays, particularly where the array is raised above the roof slope, can be very large.

Large Systems

The array mounting system must transfer loads to columns, shear walls, or ground.

Most solar systems are installed on roofs of existing, rather than new buildings. The condition of the roof must be accessed in both its structural and its sealing capabilities.

When investigating the roof's structural capabilities, the roof is inspected and structural engineering calculations are carried out. In steel structures, where the strength of the steel is unknown, the steel may be tested instead of using conservative strength estimates from the codes. This permits larger roof loadings and may reduce the quantity of framing required. It may be wise at times to confer with the structural firm who originally designed the building.

Local wind gusts or snow accumulations caused by neighbouring buildings or topography should be observed and determined as not being abnormal.

It is also important to consider potential snow drifting patterns on the roof as a result of collector placement. Work examining this problem is currently underway. It appears, as a general rule, that a clear height of at least one meter between the roof and the bottom of the collectors is required.

It is important to assess the condition of an existing roof before solar collectors are installed. If the roof was in poor condition before the collectors were installed, the contractor may falsely be blamed for damaging the roof by his installation. Repairs will also be much more difficult when the array is in place.

Because any roof penetrations will risk the chance of water entering and possibly incurring damage to the building, we should check:

- Whether the roof will need repair or resurfacing in the near future; if it does, the repair work will be much easier before solar collectors are installed.

- That the roof drains adequately; it will be difficult to seal roof penetrations which are continually submerged.

- If pipe penetrations can be made at any convenient location or are they restricted to specific areas.

1.C MAINTANENCE CONSIDERATIONS

When a solar system is contemplated for a particlular application, consideration of the availability of operating personnel is important. This is a particular concern when the solar system will be the most complex system in the building, or where it will be the only sophisticated mechanical system.

Because solar systems usually have an automatically activated backup system, assessment of the solar system's operational status is not always easy. With conventional space heating or water heating systems, a failure is usually noted by the users or building occupants and quickly reported. With a solar sytem where the backup activation is automatic, the users or building occupants will not in most cases be aware of a solar system failure. In these cases it must be left up to experienced and qualified operational personnel to continually monitor the status of the system.

If the required personnel are not available, the installation of the system may be very unwise.

SUMMARY - Section 1

Evaluating opportunities for solar energy systems is an important initial step to designing a solar installation, which is cost effective, efficient, requires minimal maintenance and which is in fact necessary. The solar system designer should proceed through the following checklist to ensure that he has considered all aspects of obtaining so called "free solar heat":

good assessement of load ________
load profiles ________
can load be modified to suit solar system ________
are any energy conservation plans to be brought forth ________
smooth integration with existing heating system ________
structural adequacy of roof, if collectors are to be roof mounted ________
obtaining accurate design load data ________
collector placement (shading, vandalism, dirt) ________
maintain piping slopes ________
maintenance requirements (complexity of system) ________
space requirements (collectors and all associated mechanical and electrical equipment ________

2. PRACTICAL DESIGN CONSIDERATION

2.A LOAD EVALUATION

In order to match the load to the available solar radiation we must determine the magnitude of the load and how it varies on a daily, monthly and seasonal basis.

A laundry washing facility may require more hot water in the early part of the week rather than the latter part. A bottle washing company often requires more hot water for washing bottles in the summer months.

To measure the energy used we may check fuel or electrical billings. When these are not specific enough we may want to monitor the load. A combination of the following instruments may be used:

- Kilowatt hour meters may be used to measure electrical comsumption; a single meter may be read for total electrical usage or individual meters may be installed to determine individual equipment electrical usage.

- Flow meters may be used to measure the fuel consumption or quantity of heated water which is used.

- Thermometers tell us the flow temperatures and give us an added indication of the amount of energy required.

- An energy meter, looking at the hot and cold water temperature, as well as the flow rate, can be installed.

The data may be manually or automatically recorded. The collected information (temperature, flows and energy consumption) is transformed as a load profile which outlines daily, weekly or monthly peaks.

With this information the feasibility for a solar system may be judged. The contribution of a solar system may be increased if the load can be altered to suit the collected energy patterns.

In many cases, patterns for energy consumption in a process are set by historical considerations and are not necessarily fixed. As an example, some plants operated at peak load during night time hours to obtain preferential off peak electrical rates. With the introduction of a solar system, it may be advantageous to shift peak operations to daytime periods, dramatically reducing solar storage requirements.

In other cases energy may be required for infrequent batch processing of items. If some latitude in timing is available, the effectiveness of the solar system can be dramatically increased if this batch processing is coordinated with available solar radiation.

2.B SYSTEM TYPE CONSIDERATIONS

There are two main types of solar systems; Air systems and Liquid systems.

Air systems require the movement of large volumes of air between components, comparatively bulky storage media and large air handling units. These make space requirements, especially in a retrofit situation, difficult to meet. Air systems can be expensive to install because of the effort required to seal and insulate ducting and components.

Liquid systems are more common in the larger installations we are most interested in. There are a number of different types of liquid systems, usually classifed by the freeze protection method employed.

- Closed loop

- Drain back

- Drain down

Each type of system has some inherent differences, requirements and cautions in design.

Closed loop systems use a glycol solution for the heat transfer fluid. Ethylene glycol is toxic and precautions, such as double wall heat exchangers, must be used to ensure that potable water is not contaminated by fluid leaks.

Propylene glycol is more expensive and is less toxic, but many local codes require the same precautions as with ethylene glycol.

Closed loop systems require facilities to fill the system and bleed off any air in the system. This is often neglected in the design.

With closed loop systems the fluid remains in the collectors at all times. In periods of high solar availability and low load the collectors can stagnate. The high temperatures can boil off the fluid and cause a breakdown of the glycol which results in an acidic corrosive fluid. Therefore, closed loop systems should have some method of protecting against over temperature. This is usually done by having a heat rejection coil or dumping of heated water to create an artificial load.

Pipes and valves should be arranged so that the collectors cannot be isolated without provision for pressure relief. Stagnation under these conditions could result in a dangerous pressure buildup.

The properties of glycol requires special attention in piping methods. Glycol will leak where water will not. This often requires special seals for items such as heat exchangers and coils. Glycol is not compatible with some materials so all components must be specified as suitable for glycol service. Most expansion tanks specified for use in PUSH solar heating systems have butyl rubber diaphrams that decompose over time due to chemical reaction with propylene glycol. With propylene glycol EPDM diaphrams must be used. Glycol also expands significantly over the temperature ranges in solar systems so an expansion tank of some sort is required.

Drain down and drain back systems require special attention to their piping design to ensure quick and complete draining of the system.

Drain back systems require pumps of higher head capabilities than closed loop systems to overcome the static head of the system before the establishment of the syphon effect.

Multibank drain back systems exhibit another common problem. If a syphon is established in one or more banks before others, the pump may not be able to fill the other banks. This problem is usually combated by the use of oversize returns so a syphon cannot be established or by restricting the return lines of individual banks. Automatic flow setters on the return lines at each bank accomplish this and simplify flow balancing of the array.

Drain down systems have fallen into disfavour in Canada because of poor experience with the reliability of the drain down controls. This has resulted in freeze up of a number of large systems, causing extensive and expensive damage to the arrays. If a drain down system is proposed, failsafe freeze protection controls are an absolute necessity.

2.C ARRAY MOUNTING

2.C.1 Types of Mounting

Numerous methods are available for mounting collector array. These vary with the choosen location for the array, either ground, wall or roof mounted.

In this section we will deal mainly with the design considerations for roof mounted arrays as these are by far the most common and present the greatest problems.

If the roof to which the collectors are to be mounted is flat, the collectors will have to be tilted at an angle to the roof. If the roof is sloped the option exists for either mounting the collectors parallel to the roof or sloping them at an angle to the roof.

Arrays Mounted Directly on a Sloped Roof

If possible, mounting the collector array on a sloped roof is usually the simplest and cheapest mounting method. The optimum slope for the collectors is determined primarily by the type of load and latitude of the location. The collector performance is reduced when the slope is off optimum but by very little if the difference is less than fifteen degrees. It is often much cheaper to mount an array on a sloped roof, and if necessary, to use extra collectors to make up the performance difference due to off optimum conditions than to incur the expense of a mounting frame.

Direct mounted collectors are best installed away from roof edges to :

- facilitate installation

- reduce wind loads and the danger of falling snow and ice

- facilitate piping

There should be drainage beneath collectors. A minimum of 50 mm allows for drainage and ventilation beneath the collectors. This limits the possibility of water ponding and fungus growth.

For arrays mounted directly on a pitched roof, the collectors can be mounted anywhere on the roof, if the collector and hardware weight is small as compared to design snow loads. This is usually the case in Canada.

Examples of Mounting Methods

CASE 1. Truss of Rafter Roof (which has access to attic)

- spanner mounting is recommended when attic is accessible.
- in this case collectors are usually placed high up on roof to allow working space inside the attic.

CASE 2. Joist Roofs and Cathedral Ceiling

- collectors can be lag bolted to joists.
- the location of the array on the roof will be influenced by piping runs, ease of fastening, and asthetics.

The lag screws must be mounted in pre-drilled holes in the centre of the structural timbers. As least 50 mm (2") of the lag screw's thread must penetrate into the timber.

Finding the joists can be a problem. The most common method is to drive small nails through the sheathing until both edges of the timber are located and to measure to the centre of the joist.. The holes can be sealed with silicone sealant or equivalent. The lag screws must also be well sealed with silicone in the pre-drilled holes and over the head of the screws.

CASE 2. Alternate

A modification of the spanner mounting method may be used on cathedral ceiling if one is willing to break into the finished ceiling to secure the threaded rods and then repair the ceiling.

Raised Mounted Arrays

Raised mounted arrays can collect snow and are subject to high wind forces; these forces are transferred down the support frame and applied as point loads to the roof structure.

Collectors are raised off the roof by a support system in order to give them an optimum tilt angle.

On smaller and residential systems mounting racks are placed directly on the roof. On larger projects, framing members span the roof and transfer loads to the roof support points. The collector mounting racks are then attached to this framing. On these projects, roof inspections and collector mounting designs are usually done by a structural engineer.

Smaller residential projects may be designed by specially prepared handbooks or come as packaged systems.

No attempt will be made here to design collector racks, but rather, to show attachment methods to roofs and mounting methods so as not to damage the roof membrane. Although structural considerations are very important, these are common knowledge to qualified people, whereas adequate methods of penetrating, fastening to, and sealing roofs with collectors are not.

It is essential to know how roofs behave and how they are built up before designing a mounting method. The condition of an existing roof must be assessed prior to the solar system installation so that faults which develop in the roof are not automatically blamed on a bad installation. If the roof must be repaired, do it before installing the array.

Problems are likely to develop on roofs that do not drain or roofs in which drainage is impeded by curb mounts for collectors.

Adequate clearance must be provided between collector rows and between the bottom of the collectors and the roof for future maintenance or resurfacing.

The following are not recommended in the installation:

- pitch pots because they require maintenance.

- eye bolts fastened directly to the roof decks.

- use of parapets as collector supports.

CASE 1.

Raised mounted array fastened to pitched roof by spanner mount method or by lag bolting to roof joists.

CASE 2.

Raised mounted collectors on flat built up roofs (curb support) as in Figure 2.C.1.1.

Note:

1. Aggregate removed from membrane
2. The membrane is cut open and the insulation is inspected
3. Because loads cannot be borne by the insulation, wood blocking is placed on the structural slab to support the curbs.
4. Roof membrane is nailed to blocking and curbs are installed
5. Curbs are anchored to slab.
6. Wood cant strips are placed alongside mounting blocks to provide a base for a bitumous flashing which will seal the roof membrane.
7. The top of curb is flashed and aggregate is placed around the mounting block to provide U.V. protection for flashing and roof membrane.
8. CAUTION: Long curbs placed at right angles to roof slopes may act as dams and prevent water from reaching the roof drains. Curbs should be divided into sections no longer than one metre.

CASE 3.

Column supports - These may rest on structural members of roofs or they may pierce roof to transmit loads down to floor slabs or footings.

See Figures 2.C.1.2 and 2.C.1.3

Note:

1. Aggregate and insulation removed
2. Column welded to and bearing on structural section
3. Pitch pocket box of exterior type placed over column and flashed with bituminous flashing to roof membrane.
4. Space in pitch pocket box filled with mastic
5. Hood and flashing placed over pitch pocket box and sealed with caulking.

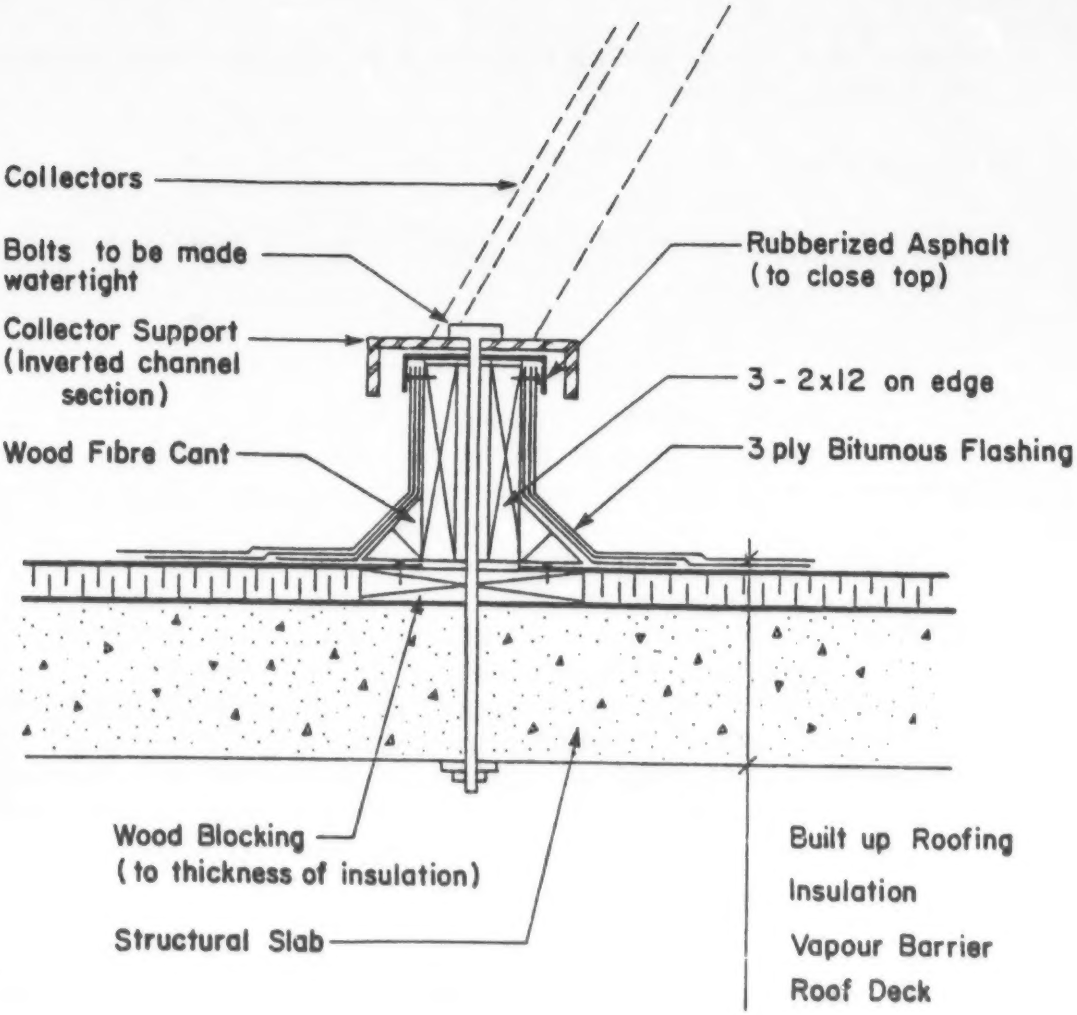
Collectors
Bolts to be made watertight
Collector Support (Inverted channel section)
Wood Fibre Cant
Rubberized Asphalt (to close top)
3 - 2x12 on edge
3 ply Bitumous Flashing
Wood Blocking (to thickness of insulation)
Structural Slab
Built up Roofing
Insulation
Vapour Barrier
Roof Deck

FIGURE 2·C·1·1
(from D.P.W. Briefing Document S-2)

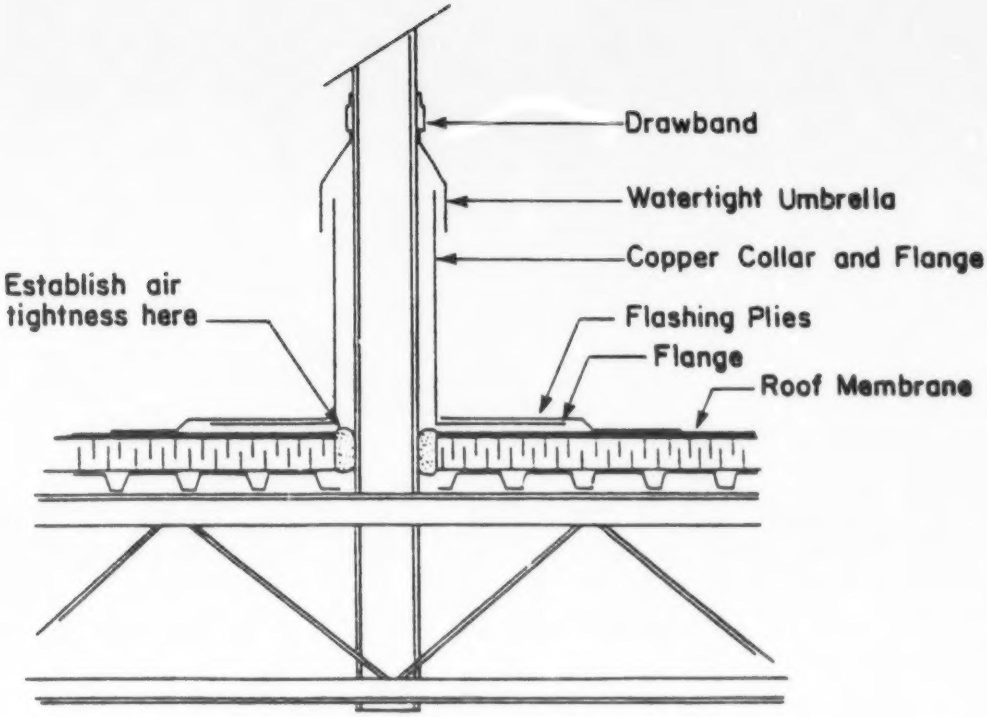
Drawband
Watertight Umbrella
Copper Collar and Flange
Establish air tightness here
Flashing Plies
Flange
Roof Membrane

FIGURE 2·C·1·2
PIPE COLUMN SUPPORT
(from D.P.W. Briefing Document S-2)

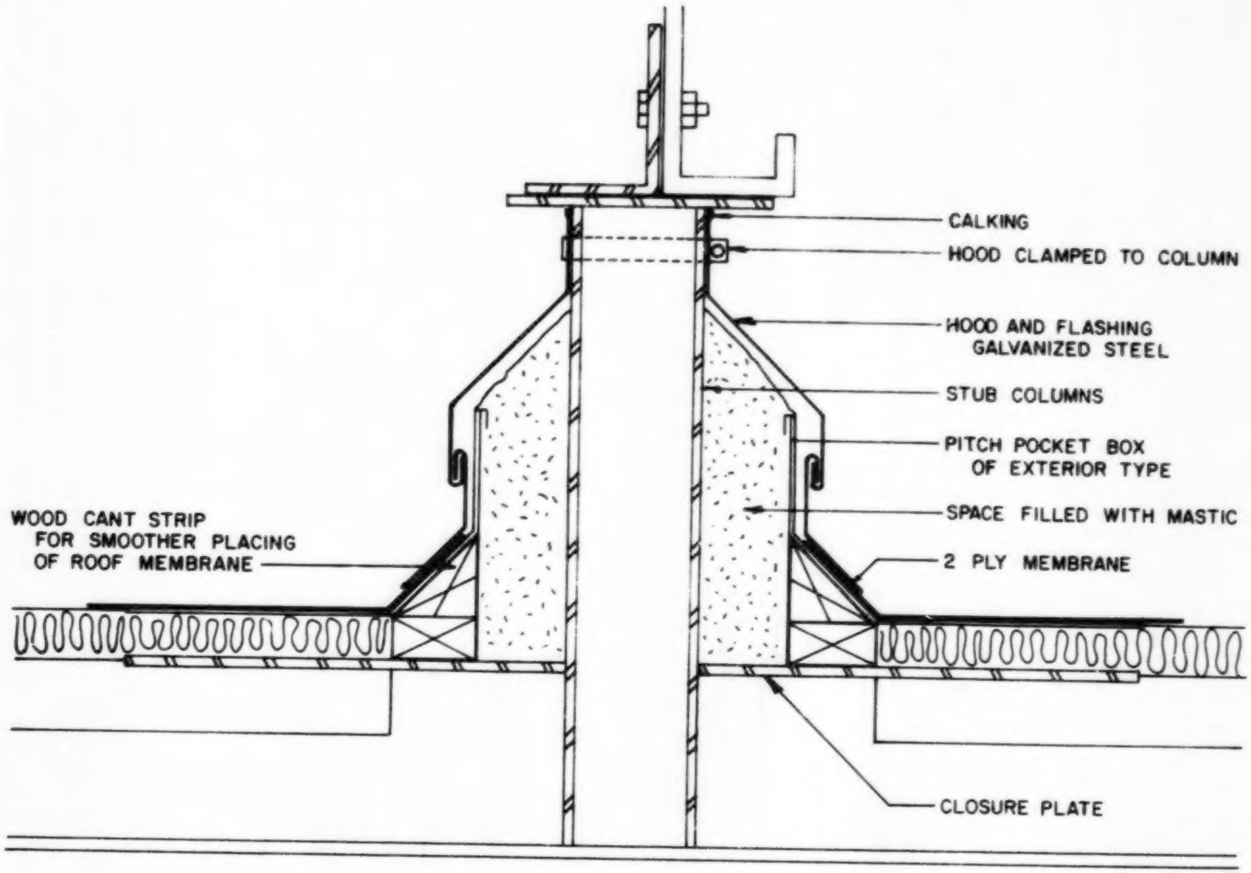
CALKING
HOOD CLAMPED TO COLUMN
HOOD AND FLASHING GALVANIZED STEEL
STUB COLUMNS
PITCH POCKET BOX OF EXTERIOR TYPE
SPACE FILLED WITH MASTIC
2 PLY MEMBRANE
WOOD CANT STRIP FOR SMOOTHER PLACING OF ROOF MEMBRANE
CLOSURE PLATE

FIGURE 2·C·1·3

2.C.2 Pipe Slope Requirements

Maintaining adequate pipe slopes is imperative for drain back systems because these rely on gravity drain back freeze protection. Glycol systems require pipe slopes for draining and flushing.

A minimum 1 in 50 pipe slope should be used on all systems. For drain down or drain back systems, this minimum slope is critical to the continued operation of the system. For closed loop glycol systems, sloped piping will allow for complete system draining, should replacement of the glycol be necessary. With large solar systems, where rows of collectors stretch for 50 metres or more, considerable height must be available beneath the collectors for pipe slopes.

Pipe mounting methods must be adequate to ensure that the pipes do not sag and reduce the original slope. This is especially true for thin wall and plastic piping materials.

2.D AIR SYSTEMS

2.D.1 Practical Aspects of Collector, Duct and Damper Sealing

In recent years the use of air type solar systems has been dramatically reduced and very little development effort is being expended on components for air systems.

Air systems, although extremely simple in concept, have proved very difficult to apply practically. Solutions to problems of collector, duct and damper leakage have proved costly to solve and these problems alone have, in many cases, negated the theoretical effectiveness of an air type solar heating system.

Some good applications for air systems do exist; however, they are very site specific. Some applications for air systems that can be considered are:

- Preheating make up air to large buildings which presently have necessary interior distribution ductwork.

- Preheating process hot air in specific commercial applications such as drying operations

- Drying of agricultural products such as grain. Although grain drying is only a seasonal application, it can be very practical in that it has a tremendously high load which utilizes every bit of energy produced from a solar system.

These application generally require a very low grade of energy.

Leakage is a major problem with most duct systems. Failure to adequately seal ducting will result in low thermal performance. For this reason, it is recommended that every duct joint be sealed with suitable tapes or sealants. Rectangular galvanized ducting used in home space-heating is not recommended as it is very difficult to seal adequately.

Some pre-insulated fiberglass ducts are available. These are generally more expensive than galvanized ducting, but save much of the installation time, and are usually much easier to seal effectively.

Sealing of the collector array is a very important consideration. This includes both connections between collectors, as well as leakage in the collector unit itself. Collectors must be carefully examined before installation to ensure that shipping has not damaged the various seals. Great care must be taken in sealing the connections between collectors.

Field experience from a number of systems has shown that:

- In order to limit leakage between collectors great care must be taken when aligning them.
- If silicone sealant is used on metal surface, better results are obtained if the surfaces are first coated with primer.
- The ducting system should be tested for leakage prior to insulating.

Glass wool and urethane insulations are the most widely available for insulating air ducts. Ducts should be insulated to at least RSI 1 (R-6) outside and RSI 0.5 (R-3) inside.

Ducting penetration into building must be done with great care to avoid both water and air leakage problems.

2.E COMPONENT SELECTION - LIQUID SYSTEMS

2.E.1 Heat Exchangers

Heat exchangers for liquid systems transfer the heat from the liquid in the collector loop to the liquid in the storage loop. These heat exchangers must be properly sized to get good system efficiency.

Double walled heat exchangers are required with systems that use potentially toxic fluids with the domestic water supply. These heat exchangers have two walls separating the fluids, so that if either wall develops a leak, the fluid enters the cavity between the walls rather than mixing with the other fluid. These exchangers usually allow any liquid entering the cavity to drip out onto the floor, thus providing visible evidence of an internal leak. CSA Standard F-379 sets out the requirements for the type of heat exchanger that should be used for package domestic hot water heating systems. Local inspectors, however, may require additional levels of protection.

Thermosiphon and drain down systems do not use heat exchangers because the domestic or process hot water is circulated directly through the collectors. Drain back systems require only a single wall heat exchanger because both fluids are water.

Internal heat exchangers are those where a coil or tank is immersed inside a storage tank. With external heat exchangers both fluids are contained in a passage where they flow past each other in opposite directions.

There are a variety of types of heat exchangers for small solar systems:

Tube heat exchanger placed in a tank - has flexible copper pipe shaped into a coil, with or without fins and secured inside the tank.

Wrap-A-Round heat exchanger - has a tube wrapped around and bonded to the outside of the tank. This type of heat exchanger is effectively double walled and is sometimes used on closed loop systems with potable water in the tank.

Parallel tube heat exchanger - consists of parallel tubes in continuous contact with each other (soldered together) where water and heat transfer fluid flow in different (counterflow) directions.

Tank in tank heat exchanger - has a single wall tank placed inside an outer storage tank. The heat transfer fluid is in the outer tank and, as an option, the inner tank may come with a coil wrapped around it.

On large systems one finds:

Tube in tube heat exhanger - has a tube fitted inside a tube (both tubes are copper). The heat transfer fluid flows in the inner tube which can be double walled with leak detection passages. Water flows in the outer tube. Fluid flow is counterflow. Sizing of heat exchanger is accomplished by adding heat exchangers in series.

Tube in shell heat exchanger - has an outer casing or shell surrounding a bundle of tubes. These are single wall types which are usually used with drain back systems. In many cases the tube bundle is placed directly into the storage tank through a port in the side.

Plate type heat exchangers - has counterflow fluid flow through heat exchanger plates. Plate type heat exchangers may be easily custom fitted to the flow requirements by adding or removing plates.

Heat Exchanger Advantages/Disadvantages

1. External heat exchangers which are not immersed in any fluid are easier to inspect and service.

2. Double walled heat exchangers are more expensive.

3. Custom made heat exchangers or heat exchangers which require special metals require long delivery times.

4. Plate type heat exchangers are expandable should the system be enlarged.

2.E.2 Pipe Systems

The piping system discussed here includes not only the piping itself but the various other associated components found in the system such as valves, expansion tanks and pipe hangers not covered elsewhere.

Piping for a solar heating system can account for a major part of the installation cost. Although most solar systems still use either copper or iron pipe, an increasing number are being constructed using plastic piping materials. This practice is particularly prevalent in small domestic solar water heating systems.

Piping runs, both inside and out, should be made as inconspicuous as possible so that the finished installation has a good appearance.

Collector piping should be kept as short as possible. This is usually possible when the collectors are centrally located to the system operating room. Long runs of piping are subject to damage, increased heat loss and pressure drops which require larger pipe sizes.

Piping should be carried out to ensure tight joints and no chance of corrosion at the joints.

- If different metals are to be used, isolate them with dielectric couplings.
- Use materials compatible with the working field. As an example, don't use zinc galvanized piping to carry glycol solutions.

Straight pipe runs longer than 25 metres should have expansion joints to prevent strain on the joints. Normally, bends and elbows are adequate to take up the expansion.

A minimum pipe slope of 1 in 50 must be used on all systems to allow acceptable draining. Plastic pipe should be sloped 1 in 30 because of the potential for sagging with time. It is good practice to install plastic pipe with continuous support over its entire length.

Ensure that the required valves and safety relief devices are included in the design. The following valves and safety devices are normally used in piping systems:

- expansion tank (closed loop)
- pressure relief valves

- air vents
- vacuum breakers (drain back)
- check valves
- strainers
- filling assembly valve
- flow control
- tempering valve

When incorporating valves into the system design, be sure that no portion of the system, such as storage tank or a row of collectors, can be isolated from the pressure relief valve intended to protect it. In the case of collectors, automatic air vents can not allow vapour to escape at sufficient a rate to protect a bank of collectors in the event it is isolated at its inlet and outlet.

In the specification of piping systems an area often overlooked is the suspension of the piping. Improperly suspended exterior piping has led to a significant number of solar system failures.

It is not sufficient to leave the design of the pipe hanging system to the imagination of the installation contractor.

Hangers should be of a standard type with proper shield or solid inserts to protect the insulation. Threaded rod used for suspending pipes must never be bent.

Pipe hangers should be spaced using the plumbing code requirements as minimum. Care must be taken to adjust hangers so that consistant slopes are maintained. Where pipes are suspended from points which might incur some differential movement, additional allowance must be made for this movement.

2.E.3 Types Of Piping

The actual selection of materials, fittings and components is dictated by the local plumbing codes. In selecting piping materials we must consider:

- the temperature extremes to which the piping will be exposed; (this will most affect solder joints and plastic piping)
- the vulnerability of piping in outdoor areas;
- the fluid that is used (some fluids react chemically with certain types of piping)

Copper Piping Systems

There are three basic types of copper piping used in domestic plumbing systems:

Type M - lightest copper pipe currently acceptable for use on solar systems; common domestic plumbing situations.

Type L - thicker wall than type M; also common in domestic water plumbing;

Type K - thick-walled for use on high pressure applications; underground piping must be Type K.

Some effort is being made to persuade inspection authorities that, because of the low pressures encountered in most types of solar heating systems, the use of very thin walled copper piping is acceptable. The type of piping being suggested is DWV piping intended for water drainage services. If this type of piping is found to be acceptable, the overall cost of piping systems could be somewhat reduced.

The use of flexible copper pipe is not recommended because of the difficulty in ensuring adequate slopes on the piping, and the poor appearance usually associated with flexible copper piping installations.

Type M, L and K hard copper piping is acceptable in most jurisdictions for use in hot water piping systems. Type K, which has a greater wall thickness, is recommended in solar heating systems where the piping may be subjected to physical damage, such as with ground-mounted collector arrays.

Galvanized Threaded Pipe

Galvanized threaded pipe may be used with water or process hot water. Teflon tape should be used with all pipe fittings. Corrosion protection (gaskets) should be provided at connections to copper or other pipe. Water will eventually corrode galvanized pipe. Galvanized pipe must not be used with glycol solutions.

Black steel piping

Black steel piping is used on glycol loops. Where large runs of pipe are to be used it is cheaper than copper pipe. Pipe fittings .should use teflon tape and be protected against corrosion at the connection of dissimilar metals.

Stainless steel pipe

Stainless steel pipe is used with corrosive fluids or where conditions dictate special contamination considerations.

Flexible epoxy fiberglass

Flexible epoxy fiberglass is used beneath roadways where there is a danger that other pipes may break.

Plastic Piping Systems

The use of plastic piping is, in cases, restricted by local fire regulations which do not allow plastic materials in commercial or industrial buildings. Where allowed, two types of plastic piping have been found acceptable. These are CPVC and Polybutalene.

CPVC piping is joined with a solvent weld, threading, or mechancial couplings. When using CPVC pipe in solar systems only solvent welded joints are recommended. Only special CPVC cleaner solvent and cement can be used. Note that the solvents and cements used with ABS and PVC are not suitable for CPVC piping. Joints made with inappropriate solvents and cements will likely leak or fail entirely. CPVC will sag significantly at about 70°C, and must be supported every 1 metre where it runs horizontally.

Where **polybutylene piping (PB)** is used in the vicinity of copper piping, care must be taken to ensure soldering paste does not come in contact with the plastic piping because it causes severe deterioration of the plastic compression fittings. PB piping also has a minimum bending radius which must be observed during installation to prevent the pipe from collapsing.

Polybutylene pipe systems can be joined with insert compression fittings or with heat-of-fusion techniques. When using PB piping with solar systems, only compression type fittings are recommended. Some solar manufacturers provide pre-insulated PB pipe for solar installations. PB piping will begin to sag at about 92°C.

2.E.4 Collector Interconnection

In the area of liquid solar heating systems a number of developments have taken place. Single header collectors have been developed both in the form of flat plate collectors and evacuated tube collectors and these types of designs have greatly simplified the installation of solar collector arrays.

Collector interconnections have presented numerous problems and no one type of interconnection system has been immune. Interconnection can be classified as expansion or non expansion type and as solid or flexible.

The various collector manufacturers specify the maximum number of collectors that can be coulpled is series without the use of some expansion device.

Collector interconnections must be able to:

- take up expansion and contraction between collectors while minimizing stresses in the couplings.

- prevent leakage at collector couplings

- prevent galvanic corrosion

The materials used for collector interconnections must be:

- durable

- resist weathering and sunlight

- compatible with heat transfer fluid

Copper Pipe Connections

Because of high temperatures in the vicinity of collectors, 95/5 tin/antimony solder should be used.

Where more than six or seven collectors, or as specified by the collector manufacturer, are mounted together, some provision for thermal expansion must be made. This limit prevents expansion and contraction from putting too much strain on the joints and sealing materials.

The major disadvantages of this type of connection are the precise alignment requirements and the difficulty in removing a single collector at a later date for repair.

Hose Connections

Some non-metallic hose connections are made with:

- EPDM

- VITON (fluoroelastomer)

- SILICONE

A preferred connection used with internal headers is silicone hose held in place with silicone caulking and stainless steel gear clamps. It is preferrable to use clamps on pipes which have ferrules. The type of connection can easily be damaged by improper tighting and often will require several retightenings before a perfect seal is obtained.

Victaulic Couplers

Used for connecting internal headers, they have EPDM gaskets and temperature ranges up to 170°C (Victaulic 77). Expansion bellows are often used with victaulic couplings. This type of coupling requires precise alignment of the headers, but allows for easy replacement of collectors. This type of interconnection is difficult to insulate.

2.E.5 Building Envelope Penetrations

Solar piping may enter a building in many different locations. Roof penetrations are very common and wall penetration may be above or below grade.

All penetrations require a tight and lasting seal. It is important that pipes be restrained firmly where they enter and that the space around the pipes be well sealed.

When penetrating a shingled roof, aluminum, rubber flashing flanges or boots are used, as illustrated in Figure 2.E.5.1. The flange is sealed at the pipe with a neoprene collar. The pipe insulation stops at the flange collar and does not penetrate through the building. This prevents any water from entering the building from beneath the insulation material. At the roof level the flange is mounted with the upper portion placed beneath and the lower portion above the shingles.

When penetrating flat roofs, pipes may be sealed inside of pitch pocket boxes as shown in Figure 2.E.5.2. The pipe is sealed inside the box with a sealing compound and caulking. The exterior of the pitch pocket box is sealed to the roof with a bituminous membrane and flashing.

Wall penetrations must maintain the minimum horizontal pipe slopes. A separate hole a few mm larger than the pipe is required for each penetration. Pipes are sealed to the wall with caulking as shown in Figure 2.E.5.3.

Underground pipes, because they need protection against ground settlement at the wall, should be inserted in a steel pipe, which also passes through the wall and extends for a few feet on the other side. The sealing is accomplished with caulking.

2.E.6 Underground Pipe Runs

Underground pipe runs are expensive, cannot be routinely inspected and are to be avoided except for circumstances where they are absolutely necessary.

If copper pipe is used, underground piping must be of type K. Piping should be buried beneath the frost line and properly bedded to prevent rocks or other objects from puncturing the pipe or insulation and to prevent settlement.

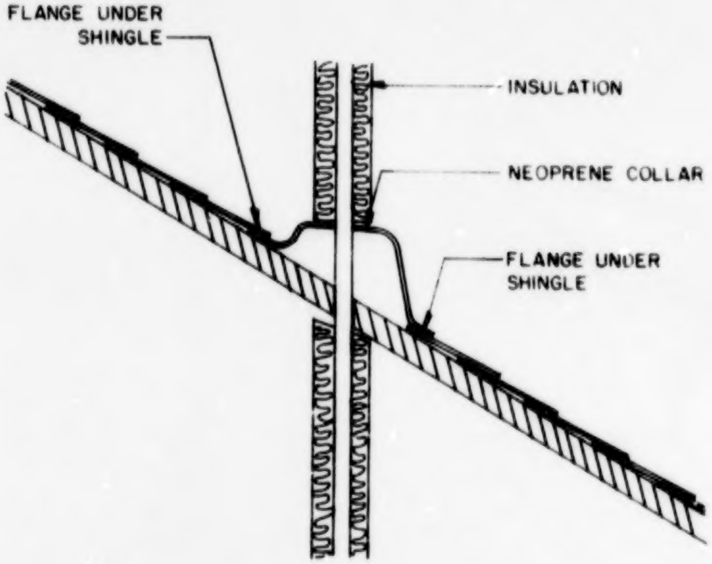
FLANGE UNDER SHINGLE
INSULATION
NEOPRENE COLLAR
FLANGE UNDER SHINGLE

FIGURE 2·E·5·1
PIPING ROOF PENETRATION

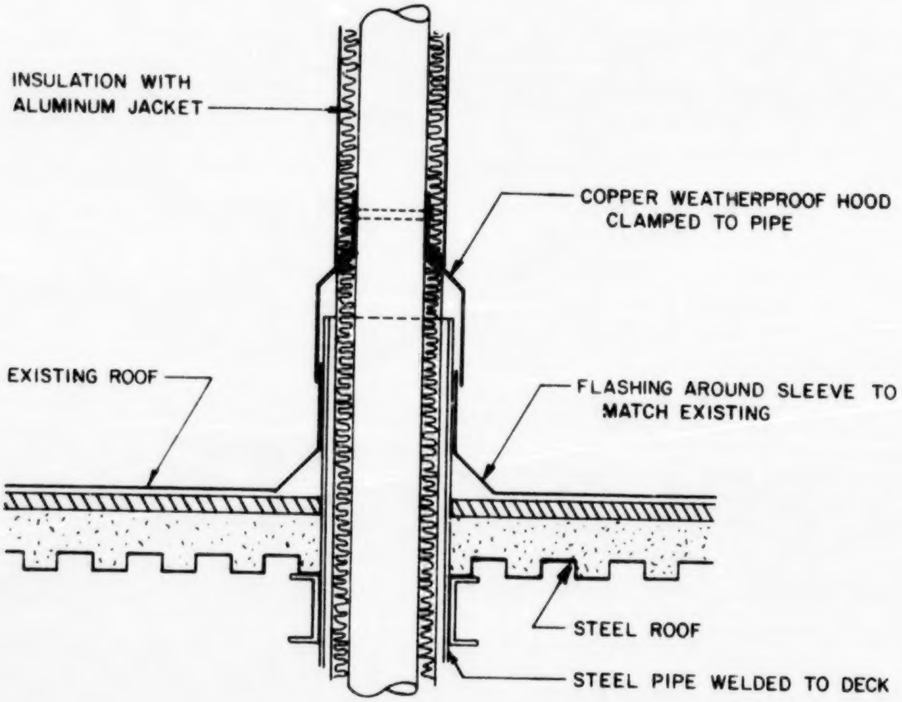
INSULATION WITH ALUMINUM JACKET
COPPER WEATHERPROOF HOOD CLAMPED TO PIPE
EXISTING ROOF
FLASHING AROUND SLEEVE TO MATCH EXISTING
STEEL ROOF
STEEL PIPE WELDED TO DECK

FIGURE 2·E·5·2
PIPING ROOF PENETRATIONS

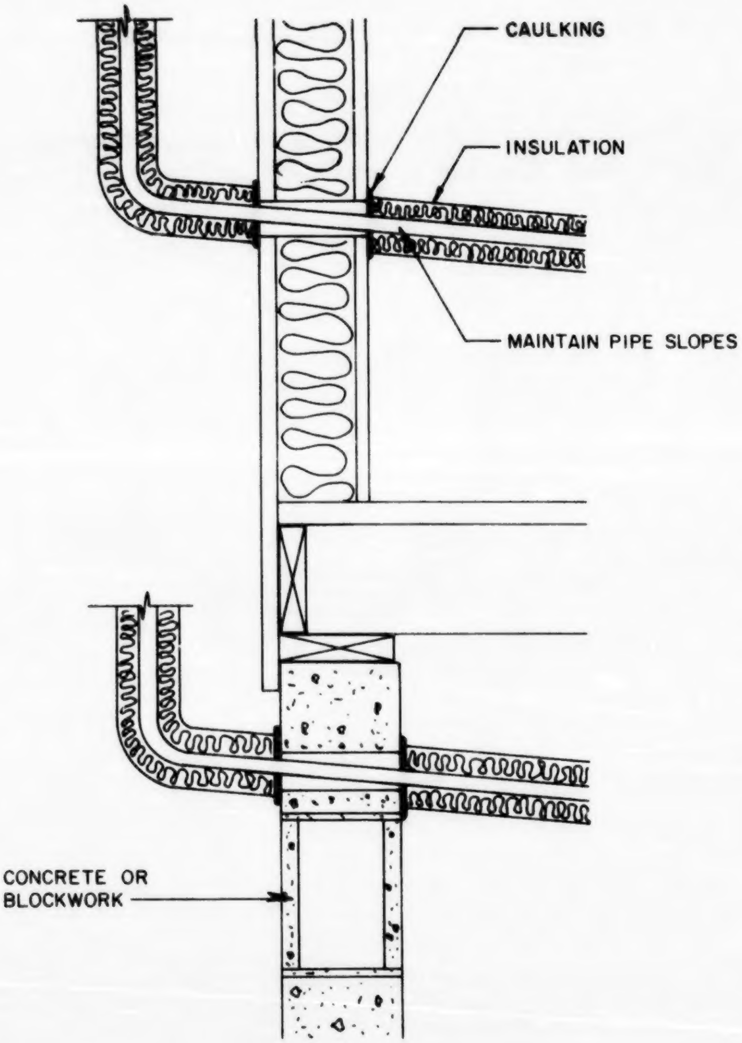
CAULKING
INSULATION
MAINTAIN PIPE SLOPES
CONCRETE OR
BLOCKWORK

FIGURE 2·E·5·3
PIPING WALL PENETRATIONS

The recommended minimum insulation level is RS1-1 (R-6). It is important that the insulation remain dry. The insulation should contain as few joints as possible.

Where piping runs through or under a foundation it should pass through in iron pipe sleeves, with the sleeve filled with sealant.

2.E.7 Insulation Requirements

The insulation of solar heating systems is an extremely important consideration in the installation procedure. Without proper care in the insulation of piping and components in a solar heating system, many of the benefits of an otherwise well designed and installed system will be lost.

The insulation of a solar heating system also goes a long way to producing either an acceptable looking installation or an eyesore. In order to produce an acceptable looking result, insulation must be selected and applied with care, both inside and out.

When insulating a solar system the storage tanks are an important consideration. The overall heat loss from a poorly insulated solar storage tank can be very significant. Insulation methods, standard in the mechanical industry, are generally acceptable in this area.

Selection of Insulation Materials

There are two major considerations when selecting insulation materials for solar applications. These are:

- proper insulation levels,
- conditions to which the insulation will be subjected.

Levels of Insulation

The selection of the proper levels of insulation for the various components of a solar heating system must be based mainly on an economic evaluation. As levels of insulation are increased, not only does the cost of the actual insulating material increase but also, there are increased costs for insulation jacketing, pipe hangers and installation. As a minimum requirement the following insulation levels should be used:

- RSI-1.0 (R-6) for exterior piping,
- RSI-0.5 (R-3) for interior piping.

Insulating Materials

When selecting insulating materials, particularly for interior use, code requirements for flamability, flame spread and smoke generation, must be considered.

Numerous insulating materials are available for use on the piping of solar heating systems. The most commonly available materials are:

- Glass fibre,
- Urethane or isocyanurate foam,
- Elastomeric,
- Polyethylene.

A major criteria for the selection of insulating materials is the conditions to which it will be exposed.

Glass fiber insulation is very susceptible to absorption of moisture and, hence, must be extremely well sealed in outdoor applications. It is bulky in comparision with the alternatives. This type of insulation should not be used on the exterior portion of a solar system.

Urethane and isocyanurate insulation are not susceptible to moisture absorbtion and are less bulky than glass fiber insulation but are relatively expensive. These materials are the best choices for exterior use. Special premolded corners and tee sections are available from most manufacturers.

Elastomeric insulation has a relatively high thermal resistance for its thickness and does not absorb water; however, it is susceptible to deterioration from ultra-violet radiation. Protective paints are available but they must be applied frequently and have not, in many cases, been proven effective. This type of insulation can be applied to the piping by slipping it on before soldering the pipes together. It must simply be pushed away from the joints being heated. Elastomeric insulation tends to shrink with age; therefore, it should be compressed by at least 5% during installation.

Polyethlene insulations are ultra-violet resistant, do not absorb moisture and are relatively inexpensive; however, they are not intended for use above 100°C and, hence, are not suitable for use near collectors which may stagnate.

It is advantageous from both cost and durability standpoints to consider using different types of insulation for interior and exterior applications.

Weatherproofing and Protection

The protection of insulation from degradation has several facets:

- protection from water absorbtion,
- protection from ultraviolet deterioration,
- protection from physical damage.

As discussed earlier, some materials are subject to either of the first two problems while, in general, all materials are subject to the third problem.

If non-waterproof insulation is used in outdoor applications, it must be extremely well jacketed if it is to be of any real long term value. The following rules should be observed:

- all seams must point downward,
- all exposed devices, other than those that must be uprignt for proper operation, such as automatic airvents, must project downward,
- all joints in the jacketing must be sealed with a sealer and/or tape approved for the purpose. It is best to obtain the tape supplied by the jacket manufacturer for the purpose. Duct tapes or electrical tapes are are not acceptable.
- termination of insulation at the connection to collectors must be completely sealed to both the outer insulation jacket and the pipe itself. Sealants used for these purposes must be capable of withstanding the extreme temperatures of stagnation that might be encountered.

For these reasons, a water absorbing insulation, such as fiberglass, should not be selected for exterior use. Urethane and isocyanurate insulations have proven to be best for exterior application because of their low water absorbing characteristics.

Most insulating materials are acceptable for indoor applications with the major criteria for the installation of all types being the same - completeness and neatness. The fire ratings of insulation such as elastomeric type, must, however, be considered.

Insure that all piping carrying hot water is insulated, including any piping between the solar storage tank and the auxiliary storage tank. All valves and test points installed in these lines should also be insulated.

Miter all angled joints in the insulating material if preformed elbows and tees are not used. Tape all joints with a tape of the same colour as the insulation jacket and in no case use electrical or masking tape for this purpose. Tapes are available for use with most types of jackets from the the manufacturer. The manufacturer can be consulted for his recommendations.

Insulation Safety Considerations

The basic purpose of pipe insulation is to reduce heat loss and, hence, increase the efficiency of the solar system. Another purpose that is also of great importance is protecting people from burns.

It must be remembered that solar heating systems can produce water of very high temperature. Water or fluid of this temperature will be circulated, at times, through the loop between the solar storage tank and the collectors. Depending on the plumbing arrangement of the system, high temperature water may also travel from the storage tank to the tempering valve where it is mixed with cold water.

All piping which can potentially be heated to high temperatures must be completely insulated. Valves or other fittings that project through the insulation in these portions of the piping must also covered if they are in areas where they might accidently be touched.

2.E.8 Pumps

Pumps for drain back systems must abe sized with significant capacity to reliably fill the system and establish full circulation. wht the pump drawing from a non-pressurized tank, the minimum suction head requirements for the pump must be considered.

Pumps circulating potable or non-treated water must be either bronze or stainless steel. When glycol is used the pump must be specified with seals and gaskets of suitable materials.

2.F CONTROL SYSTEM STRATEGIES AND TRADEOFFS

The purpose of the controls in a solar heating system is to regulate flows through the various loops of the system in such a way that efficiencies are optimized. A second and equally important function of the controls is to protect the various system components from harmful temperature extremes.

The control logic should be devised to perform these functions in the simpliest fashion possible. Generally, the simpler the controls the more likely they will continue to operate properly and be maintained on site.

The failsafe aspect of control systems is extremely important. Once a design is devised it should be carefully examined to ensure that no simple component failures could cause a drain back or drain down system to stay full in below freezing conditions or for a glycol system to continue to circulate sub freezing glycol through a heat exchanger.

After failsafe operations have been examined, the system must be examined to ensure that, in the event of a primary control failure, the failsafe backup feature will not simply operate the system until the backup control system also fails, without giving the required alarm.

Operation and Options Available on Differential Controllers

Control systems for solar heating systems have changed little in a conceptual way from those developed for the earliest applications. Most solar heating systems are controlled by a differential temperature controller, usually looking at the temperature difference between the solar collectors and the storage system. Great advances have, however, been made in the reliability and the maintainability of control systems.

Numerous control systems now available include diagnostic packages and some are available complete with data acquisition systems which record critical temperatures, flow rates and energy quantities automatically.

Although the more sophisticated of these controllers are only suitable, because of their cost, for large engineered solar heating systems, even the controllers available for small package solar heating systems are substantially improved over earlier models. More reliable sensors are being employed, reducing the incidents of control system failures and added features such as freeze protection and over temperature protection are available.

Sometimes controllers may be faulty, improperly located, improperly installed or incorrectly set. This hinders the operation of the entire solar system. It is important, therefore, that controllers be checked before and during commissioning of solar systems.

Experience has shown that control system problems are one of the primary sources of operation problems and poor system efficiency. Extra attention to selection, installation, set-up, checking and maintenance of control systems is well worth the effort.

The **basic on/off differential controller** has two temperature sensors. The controller starts the system whenever the temperature difference between collector and storage is greater than a preset amount and stops the system when this temperature drops below a preset amount.

The **proportional flow differential controller** operates the same as the basic type except that it can vary the pump speed. This regulates flow of fluid to the collectors to account for the amount of solar radiation available. This reduces pump cycling and improves collection efficiency.

The **differential controller with protection features** operates the same as the on/off proportional flow differential controllers except that it also has sensors to protect the system from freezing or overheating.

Some controllers behave as data acquisition systems which record other data that is used to judge how well the system is operating.

The control operation of a differential controller has the following basic sequence:

1. Pump is switched on when the collectors are warmer than the storage tank by a preset amount, usually 10°C.

2. Pump is switched off when the temperature difference between the collectors and the storage drops to another preset value, usually 1 to 3°C.

3. Most controllers have a built in "fail safe operation" that shuts the pumps off when there is a broken sensor wire. If a system failed in the "on" position, wasted energy and possibly freezing of the system would occur.

Controllers may optionally have:

1. An over temperature protection sensor located in the storage tank which turns off the pump if the storage tank temperature exceeds a preset valve, approximately 70°C.

2. Freeze protection sensors turn off the pump when temperatures in the return piping drops below approximately 5°C. This prevents drain back systems from freezing and prevents glycol systems from freezing water in the heat exchange loop.

Pumps for a secondary circulation loop where service hot water or DHW is heated by circulating it through an external heat exchanger are usually controlled by the same control as the collector pumps. The collector and secondary pumps are then turned on and off together.

Control systems must allow the overall operation to proceed simply and with little manual input. It is to be expected that if control systems are to operate the solar systems efficiently, mechanical equipment must have been well designed and sized.

2.G COMMISSIONING REQUIREMENTS

Commissioning requirements may be divided into two sections; start up and commissioning.

Start up includes a series of tests and inspections conducted by the system installers and witnessed by a representative of the owner of the system. These are performed during and immediately after construction to ensure that the system and its components are correctly installed, capable of operating at specified levels and capable of being commissioned.

Commissioning includes a number of discrete tests. The performance and successful completion of these tests ensure that the installed system meets the design specifications.

Many solar heating systems, employing relatively sound designs, have never operated as expected. In many cases this was due to lack of proper commissioning of the system at the outset. The commissioning of the solar heating system is one of the least understood and most often neglected processes in the entire solar design and construction industry. In recent times this problem has begun to be recognized and substantially more effort is being placed on ensuring that a solar heating system is actually operating as designed before it is signed off as complete.

The proper commissioning of a solar heating system not only ensures the system is operating as designed when the system is complete but allows the setting of bench marks to which system parameters can be compared over the long term and hence ensure the proper operation of the system.

The **start up** work involves the following series of inspections and tests:

1. Site Inspections

The following equipment is inspected:

- Collectors
- Piping
- Insulation
- Thermal Storage
- Electrical

This is the ensure that all equipment:

- is as specified on drawings
- conforms to manufacturers spec sheets
- is correctly installed
- is capable of operating at the specified level
- is capable of later being commissioned.

2. Verification of Glycol Properties

This ensures that the heat transfer fluid:

- is of the proper type
- has required freeze protection

Tests are performed to determine the glycol fluid's reserve alkalinity, concentration ratio and ph level.

3. Differential Controller Test

The following three tests are performed to ensure that the differential controller operates correctly:

- Fail safe operation
- Differential Setting
- Drain back test (on systems using a drain back operation for freeze protection)

4. Motor and Pump Test

These ensure that the pump is installed correctly, with the correct rotation, and its operation meets the manufacturer's specifications.

5. Leak Test

The system is pressure tested to find any leaks in the systems. Leaks should be repaired and the system retested. The system may be tested in sections as construction proceeds, however, a final pressure test of the completed system is best where possible.

6. Final Visual Inspection

During this final inspection the system is checked over to ensure that it can be started up safely. Checks are made to ensure that the system is filled with liquid (closed loop), that there is flow and that manual valves are opened in the specified pattern. Collectors are checked for outgasing which would affect the system performance.

7. Flow Balancing of Collector Loop

This involves:

- measuring collector loop flow rate

- measuring and balancing bank flow rates directly or by temperature

- adjusting final flow rates.

The commissioning work involves the following tests:

1. Verification of Flow Balance

The collector loop is checked for flow and for flow direction. The pump horse power and pressure differential is measured. With these and the manufacturer's pump curve the flow rate is checked. With the system operating the temperature rise across each collector bank is measured and if necessary the bank flow rates are adjusted.

2. Final Flow Adjustment

With the collector flow balanced, a final adjustment of the overall flow rate is made. A dated locking tag is placed on the system's throttling valve.

3. Heat Exchanger Effectiveness

The heat exchanger is checked to ensure that it conforms to specifications. Primary side and secondary side heat exchanger temperatures and flows are measured and the effectiveness calculated.

4. Preliminary Thermal Performance Test

This test assesses the operation of the solar system as a whole, and compares results with the performance claimed by collector manufacturers.

The following data is recorded:

- outdoor ambient temperatures
- collector return temperature
- collector supply temperature
- total collector loop flow

Solar insolation is usually estimated rather than measured. With this data and the density and specific heat of the heat transfer fluid we can calculate:

- the heat gain per unit collector area
- the efficiency (which may then be compared with the manufacturer's claims)

2.H OPERATIONAL MONITORING

The prime purpose of operational monitoring is to detect a decrease in the solar system performance. This is done by regularly comparing certain indicators, such as temperatures, pressures and pump run times with those specified in the design and commissioning process. A change in these indicators outside of the design range means that the system has an operational problem. Identifying operational problems or system breakdowns as soon as possible is important in reducing damage to the system, and will also maximize the energy output of the solar system. Early detection of problems will also increase the life of the system and keep operating and maintenance costs to a minimum.

Checks of instrument readings and verifications of the operation or condition of component equipment are conducted on a set of schedules as follows:

daily,

monthly,

semi-annual or yearly.

Daily inspections involve visual checks of:

- system pressures,

- pump operation in relation to weather and control modes,

- system temperatures in relation to control modes,

- status of system alarms,

- evidence of leaks or pressure relief valve activation.

These indicate whether there is an immediate malfunction or not.

Monthly checks involve a more detailed examination of items that are checked daily. The recording of the information provides an historical record of the system's performance.

Semi-annual checks involve a detailed examination of the system. Many of the tests required in the startup and commissioning phase are repeated.

Some typical semi-annual checks are:

- fluid quality checks
- control system checks for:
    - sensor mountings security
    - calibration of sensors
- check on condition of filters and strainers
- testing of valves to assure functioning
- fluid level in expansion tank
- heat exchanger effectiveness test

The system design must include sufficient instrumentation to perform the daily and monthly checks. A typical package would include:

- one pressure gauge for each pump which can measure inlet and outlet pressure selectively
- a run time meter and event counter on the collector pump(s)
- temperature gauges measuring key system temperature including collector supply and return and storage temperatures.

Continued Operation of the Solar System

Once a solar heating system has been designed, constructed and commissioned, it must then be cared for. Much more emphasis is now being placed on the proper documentation of systems and the preparation of operating manuals.

Manuals are essential to the successful operation of any mechanical system. They must clearly explain the operation of the system, contain the necessary operational checklists and detail troubleshooting procedures for when a fault is found or suspected. These requirements are really no different than for any other complex mechanical system, but have often been neglected in the solar industry.

In the past, the production of manuals was generally left to the contractor constructing the system, with varying amounts of input from the design consultant. The operations manuals for many solar systems consist merely of a collection of component brochures. Although this information is important, it is only one small part of an operations manual.

Significant improvement in operating manuals has taken place recently, mainly through efforts of the various funding agencys. As an example, solar systems now constructed under the federal government PUSH programme require that manuals for the commissioning, operating, and troubleshooting be produced as part of the documentation for funding approval.

SUMMARY - Section 2

Being practical in the design considerations implies a familiarity with solar systems; it means using experience which has been gained from past problems associated with solar installations. In this way many of the tried and tested methods do not have to be tried over again if they were unsuccessful the first time. Truly useful or accurate data on solar systems can often only be obtained from a monitoring program.

The following checklist will provide the designer with some important considerations.

Accurate measurement of load (instrumentation, past consumption)

Choice of system type

- best suited to situation
- climate
- complexity
- reliability

Mounting location (wall, ground, roof)

Mounting method (flat mounted, raised mounted)

Raised mounted considerations

- design data (wind and snow loads)
- roof repairs and or reinforcement
- expensive mounting frames bearing to walls or columns
- design of mounting frames and racks, roof penetrations

Pipe slope requirements

Component selection

- heat exchangers - size, type, costs
- piping - quantity, size, costs types, pipe connections, slopes required, heat transfer liquids, thermal expansion, valves, safety devices and control devises, hangers for pipes, aesthetics
- insulating materials - thickness, weatherproofing, fire rating

Control Systems

- protection provided
- simplicity
- reliability

Commissioning

- requirements
- commissioning documents

Operation Monitoring

- what is involved
- available documents

APPENDIX A

Product	Brand Name	Supplier
Insulation		
Glass Fiber	Micro-Lok 650 Fiberglass ASJ	Johns-Manville Fiberglass Canada
Urethane or Isocyanurate foam	Solar-7	Petro-Sun Inc. Montreal, Mississauga
Elastomeric	Armaflex Rubaflex	Armstrong
Polyethelene	Therma-cell	Nomaco Inc.
Insulation Jackets		
Aluminum		General Aluminum Supply Corporation
Isogenepak, PVC	Montreal, Ottawa	Insul-Coustics
Insulation Jacket Accessories		
Aluminum or stainless steel strapping		Insul-Coustics Montreal, Ottawa
PVC Tape		Insul-Coustics Petro-Sun Inc.
Expansion Couplings		
Stainless Steel expansion bellows		Hydro Flex Inc. Hamilton
Flexible Hoses	EPDM silicone	
Victaullic Couplings	Victaulic 75 77	Craine Canada Ltd.

Product	Brand Name	Supplier
Heat Exchanger Fluids		
Ethylene or Proplene glycol		Dow Chemicals Union Carbide Corp.
Heat Exchangers		
Tube in shell Plates		American Standard Alpha Laval Tranter Canada
Tube		Darling Duo.
Differential Controller		
Paget	CM 100	Paget Electronic Ltd Carleton Place, Ont.
Delta-T	DTT-400	Heliotrope General Spring Valley, Ca.
Independent Energy	C-30	Independent Energy Inc. East Greenway, RI
Solar Systems	DT-11H	Solarsystems Industries Ltd. Richmond, B.C.
Metal Support Systems		
Unistrut		GTE Unistrut

POOR

QUALITY

ORIGINAL